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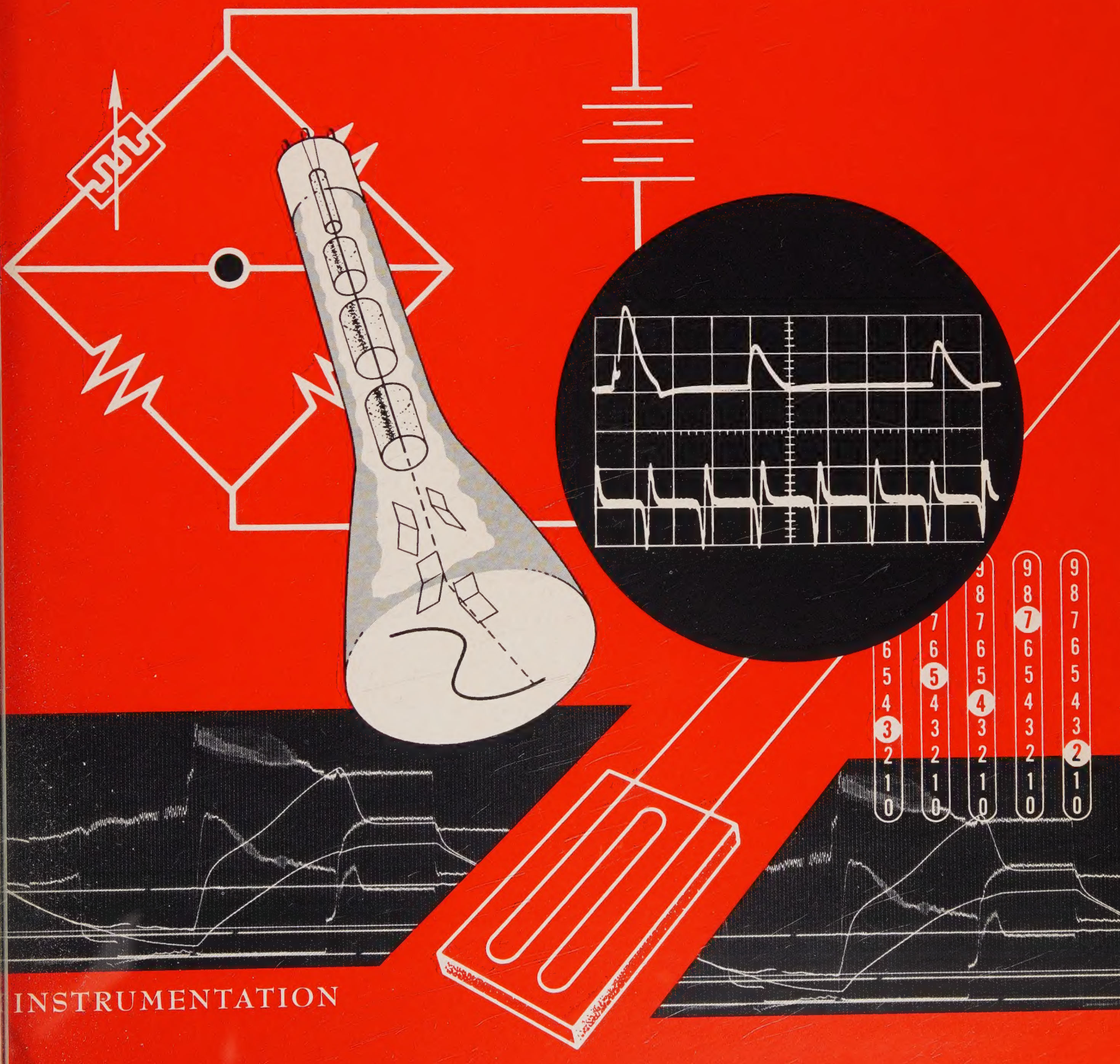
for educators in the fields of engineering and allied sciences



INSTRUMENTATION

On Being Creative

You'll never be through studying if you are an engineer—and want to stay a good engineer.

This is a message I have repeated time and time again in talks to various groups of management people.

In short, I believe that we in industry must be *perpetual students*.

But you've probably heard this before. This undoubtedly doesn't come as news to you—that you must keep on studying if you want to keep on progressing.

But there is one aspect to study that you may not have thought of.

It's the necessity—so vital in an engineer—to remain *creative*.

By creative I mean *idea-minded*.

And *open-minded!*

We have done years of work in the Division with which I am associated on the subject of "creativity."

We have studied whether people can be trained and stimulated to have more ideas.

We have found, in brief, that your creative powers, your ability to have ideas, *can* be broadened.

There are many techniques which are useful, varying from idea check lists to other principles of practical psychology.

But the overriding value of our courses and training in creativity, I am convinced, is the building of an *atmosphere of creativity* in our whole organization plus the elimination of the usual first reaction to an idea—the reaction of finding something wrong with the idea.

Gradually, everybody comes to believe that we mean what we say—that ideas *are* wanted, *are* welcome, and that every idea doesn't have to be a world-beater to be valuable.

Every idea, of course, cannot be a good one. But out of many ideas, one with tremendous potential may develop.

The main thing is to get *everybody* in the organization in the frame of mind so he is constantly searching for new and better ways of doing the job, or making the product.

One of the best habits to develop in this connection is to *receive ideas* in a *positive manner*.

Commonly, the first reaction to a new idea is to find something wrong with it.

This negative reaction protects our lazy minds from having to go further.

But the cost to the individual and to the company can be very great in ideas killed at their conception—simply because we did not explore them far enough.

Unfortunately, this habit—of negative reactions to new ideas—exists in far too many people. Engineers are no exception, especially when the idea is from a person who is less expert.

I urge young engineers, especially, to discard the habit of resisting change.

Do not—at first—be so judicial. Take the idea and see what can be done with it.

Solve the objection later.

Consider ideas that cannot be worked out as a personal failure.

Have faith that objections can be overcome.

Don't be afraid to be different.

Develop a love of study—and resolve to keep on learning.

This is the best advice I can give to young people in the engineering professions.

If you use it, I know it will help you the rest of your life.

J. A. Anderson,
Vice President
of General Motors,
General Manager of
AC Spark Plug Division

THE COVER

Man's inherent sensory systems—sight, touch, and hearing—often are limited when applied to science and engineering investigations. Instrumentation is used, therefore, to extend the human senses and provide information on both static and dynamic phenomena. In this issue's cover design, artist Richard P. Renius depicts some of the basic types of instruments widely used in research and engineering work. Shown is the cathode ray tube, the basic component of the versatile cathode ray oscilloscope; the strain gage, used in stress analysis studies; and the electronic counter, used for high speed counting. Also shown is the Wheatstone bridge, a basic circuit used in many measurement applications. Shown on the back cover is a reproduction of a multi-channel oscillographic record. Such records are obtained when information is desired on the dynamic relationships between more than one variable.

CONTENTS

PAPERS IN THIS ISSUE

CLASSROOM PROBLEMS

OTHER FEATURES

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The GENERAL MOTORS ENGINEERING JOURNAL is published quarterly by the Educational Relations Section of the Public Relations Staff, General Motors Corporation. It is intended primarily as a medium for presenting to educators in the fields of engineering and allied sciences the results of General Motors research and engineering developments in both the product and production fields.

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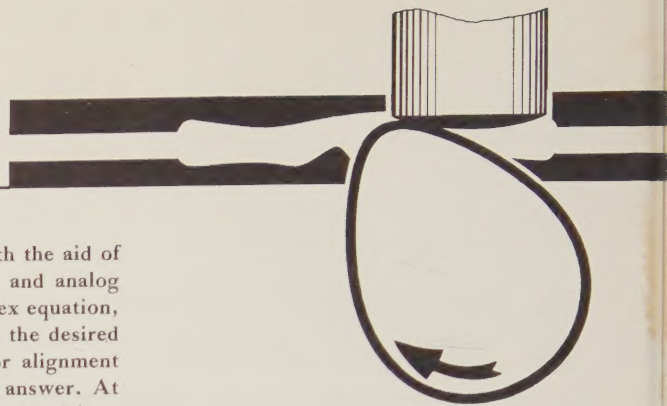
The Application of Nomograms to the Solution of an Engineering Problem



Calculations necessary to solve engineering problems can be carried out with the aid of such labor saving and time saving tools as the slide rule, desk calculator, and analog and digital computers. When it is necessary to solve a frequently used complex equation, it is sometimes desirable to have still another tool available which will give the desired answer as quickly and easily as possible. Such a tool is the nomogram, or alignment chart, which requires only the use of a straight edge to obtain the desired answer. At Diesel Equipment Division, the design of both hydraulic and mechanical valve lifters frequently requires spot checking the contact stress between the cam and lifter foot. Due to the complexity of the contact stress equation it was decided to apply nomography to simplify and expedite its solution. To chart the stress equation required breaking the complete equation into sub-equations, which were then charted individually. A series of seven nomograms was required to obtain the desired solution. To check the accuracy of the nomograms, the contact stress at a specific point in the cam cycle was calculated using both the nomograms and a digital computer. When the answers were compared, the nomographic solution had an accuracy of minus one-half of one per cent.

In the design of hydraulic and mechanical tappets (valve lifters) at Diesel Equipment Division, it is required that the contact stress between the cam and tappet foot (Fig. 1) be known in order to predict wear patterns theoretically and to determine if a wear problem is likely to exist in a given application.

During the cam lift event, from beginning of opening ramp to end of closing ramp, the contact stress varies for each increment of cam rotation. Generally, the cam lift event is specified per degree of cam rotation and may have a duration as much as 200°. For accurate wear and stress analysis, the contact stress over the complete cam lift event can be determined by computing and plotting a curve of cam rotation angle versus contact stress. Since the contact stress varies for each increment of cam rotation, computing such a curve becomes a very laborious and time consuming operation. Consequently, Diesel Equipment has complete contact stress curves computed by a digital computer.

When evaluating a design change involving an established tappet, however, it is frequently desirable to spot check the contact stress quickly at specific points in the cam cycle. The contact stress at a specific point in the cam cycle can be calculated by means of a complex stress equation* (Fig. 2) which gives the maximum contact stress between a spherical faced tappet and a taper-faced cam. Because of the time required to solve this equation by such means as a slide rule or logarithmic and trigonometric tables in conjunction with a desk calculator, nomography was applied to simplify and expedite its solution.

Various Types of Nomograms Used to Chart Equations

In general, two theoretical approaches can be used to construct nomograms—plane and analytical geometry or determinants[1]. Since the procedures for constructing a nomogram are quite detailed and are dependent on the type of equation being charted, a complete analysis of the procedures involved will not be discussed. However, a brief outline of the mechanics of constructing a nomogram will be given.

Before constructing a nomogram, the equation to be charted must be classified as to its type—for example, sum or difference of two or more variables, multiplication and/or division of two or more variables, equations having reciprocal functions, equations having more than one function of one or more variables, and in special cases, combinations of these various types.

Once the equation has been classified, the type of nomogram(s) which will represent that type of equation best then can be determined. Various types of nomograms are available—for example, those having vertical or horizontal scales with intersecting or parallel solution lines; vertical and horizontal scales with perpendicular solution lines; Z-type nomograms with single, parallel, or intersecting solution lines; nomograms having scales radiating from a common intersection point; nomograms with scales forming acute or obtuse angles; and nomograms having curved scales.

After selecting the type of nomogram to use, the next step is to establish the type of scale(s) applicable to the nomogram—for example, equally divided or logarithmic. The size of each scale is controlled by the limits of each variable. Functions, powers, and roots of variables are incorporated into the scale of each variable. Also, all constants in the equation are combined with one or several variables and are incorporated into the scale of the variable.

The final step is to construct the nomogram according to established procedures.

Nomograms Provide Specific Advantages

Since a nomogram is a graphical aid for solving an equation, it follows that a curve plotted in Cartesian coordinates also might do the job. The more complex the equation, however, the more difficult it is to obtain an accurate solution from such a curve. Also, a family of curves is required to represent an equation having more than two variables, thus impairing the accuracy of the solution by making inaccurate interpolation and extrapolation necessary.

In contrast, a single nomogram can be used to solve an equation having as many as four variables and, in special cases, equations having more than four

*The cam and tappet contact stress equation was derived in conjunction with the author's Fifth Year Project Study Report for General Motors Institute and Diesel Equipment Division. The Fifth Year Project Study is one of the requirements for a baccalaureate degree in engineering at G.M.I.

By HARVEY J. MEEUSEN
Diesel Equipment
Division

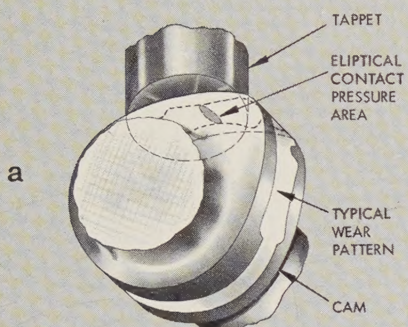
Seven nomograms solve complex cam and tappet contact stress equation

variables, with accurate interpolation and extrapolation possible. In general, however, a series of nomograms is used to solve an equation having more than four variables.

The advantages of nomographic solutions to equations can be summarized as follows:

- Once the nomogram is constructed, a solution is obtained easily using only a pair of triangles or similar straight edges
- The accuracy of a nomogram is at least equivalent to, and often more accurate than, a slide rule
- An equation can be solved with a minimum number of operations
- Repetitive calculations are not required to evaluate a given variable
- The amount and direction of change required for an independent variable in a given equation can be determined visually
- Interpolation and extrapolation can be done accurately.

The construction of a nomogram is not without its problems. Some of the problems encountered include the incompatibility of equally divided and logarithmic scales and the difficulties



P = valve gear load compressing the cam and tappet together
Z = offset of cam and tappet contact point from centerline of cam
Z_f = offset of cam and tappet contact point from centerline of tappet caused by effect of tapered cam contacting a spherical tappet face
Z_o = initial offset of cam and tappet centerline caused by stack-up of engine tolerances
s = angle of taper on cam face
Θ = incremental angle of cam rotation
R_b = cam base circle radius
R_c = cam radius of curvature at point of contact
R_f = spherical radius of tappet face
L = specified cam lift at angle Θ
L_r = cam lift of spherical faced tappet

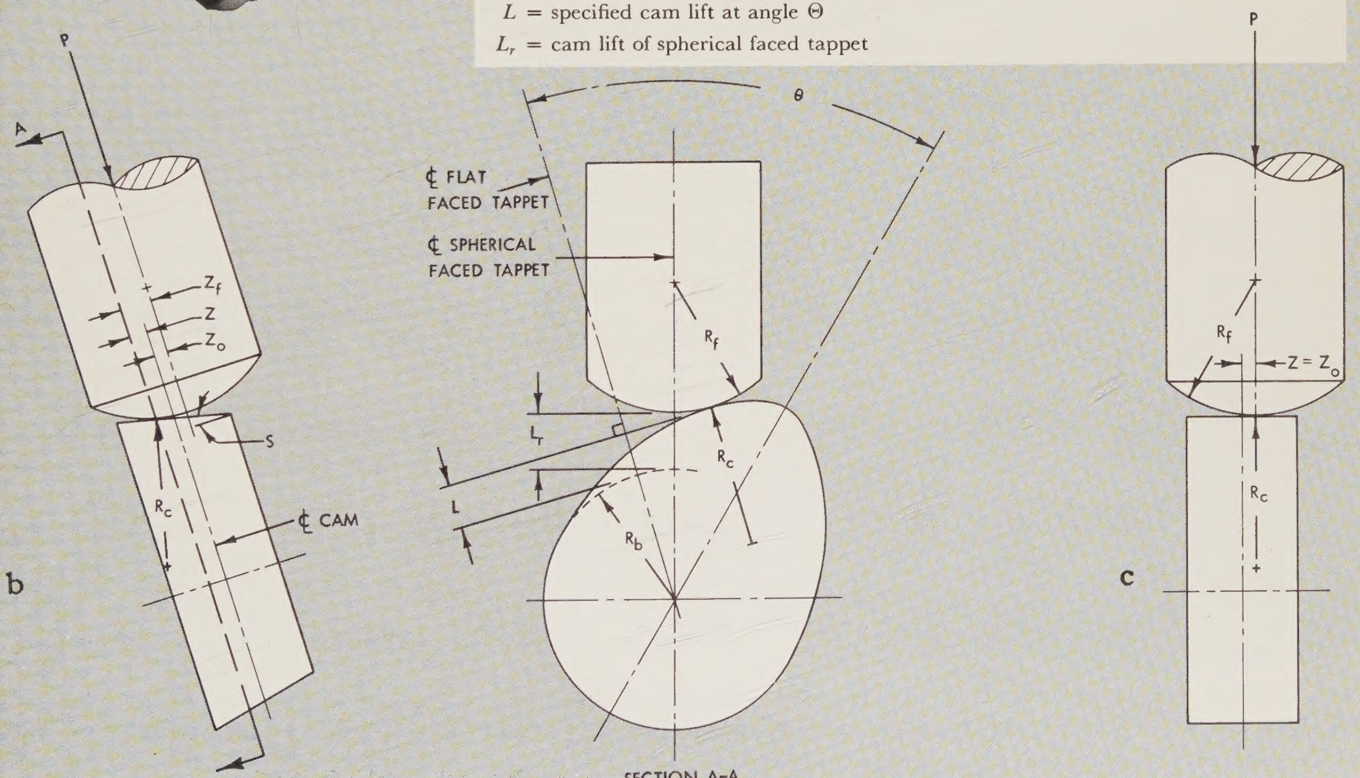


Fig. 1—Contact between a spherical faced tappet (valve lifter) and either a taper-faced cam or flat-faced cam produces an elliptical contact pressure area (a) which causes a wear pattern to be generated on the cam. When designing tappets, the contact stress between the cam and the foot of the tappet must be known to predict wear patterns and determine if a wear problem is likely to exist. To determine the contact stress at any increment of cam rotation, a cam and tappet contact stress equation (Fig. 2) is used to calculate the contact stress between a spherical faced tappet and either a taper-faced cam (b) or a flat-faced cam (c). Nomography is used to simplify and expedite the solution of the contact stress equation.

COMPLETE CONTACT STRESS EQUATION

$$S_m = \frac{\left[3\left\{\left[(RAR)^2(K_v)(L_r - VTLC) + (P_o)(RAR)\right]\left[1 + m^2\right]^{1/2}\right\}\right]^{1/3}}{2\pi XY\left[\dfrac{\dfrac{4(1-Q_1^2)}{E_1} + \dfrac{4(1-Q_2^2)}{E_2}}{8\left[\dfrac{2}{R_f} + \dfrac{1}{\left(L + mZ + R_b + 3282.8\dfrac{d^2L}{d\Theta^2}\right)\left(\dfrac{1}{1+m^2}\right)^{1/2}}\right]}\right]^{2/3}}$$

Where

X and $Y \propto \cos v$

$$\cos v = \frac{\dfrac{1}{\left(L + mZ + R_b + 3282.8\dfrac{d^2L}{d\Theta^2}\right)\left(\dfrac{1}{1+m^2}\right)^{1/2}}}{\dfrac{2}{R_f} + \dfrac{1}{\left(L + mZ + R_b + 3282.8\dfrac{d^2L}{d\Theta^2}\right)\left(\dfrac{1}{1+m^2}\right)^{1/2}}}$$

CONTACT STRESS EQUATION AS SIMPLIFIED FOR CHARTING

$$S_m = \frac{3\left[(RAR)^2(K_v)(L - VTLC) + (P_o)(RAR)\right]}{2\pi XY\left[\dfrac{3\left[(RAR)^2(K_v)(L-VTLC)+(P_o)(RAR)\right]\left[\dfrac{4(1-Q_1^2)}{E_1}+\dfrac{4(1-Q_2^2)}{E_2}\right]}{8\left[\dfrac{2}{R_f}+\dfrac{1}{L + R_b + 3282.8\dfrac{d^2L}{d\Theta^2}}\right]}\right]^{2/3}}$$

Where

X and $Y \propto \cos v$

$$\cos v = \frac{\dfrac{1}{L + R_b + 3282.8\dfrac{d^2L}{d\Theta^2}}}{\dfrac{2}{R_f} + \dfrac{1}{L + R_b + 3282.8\dfrac{d^2L}{d\Theta^2}}}$$

S_m = maximum contact stress

RAR = rocker arm ratio

K_v = rate of valve spring

$VTLC$ = valve train (gear) lash on cam side of rocker arm

P_o = initial valve spring load at installed height

m = slope of taper angle on cam face

Q_1, Q_2 = Poisson's ratio for tappet face and cam material, respectively

E_1, E_2 = modulus of elasticity for tappet face and cam material, respectively

A_1, A_2 = elastic constants of tappet face and cam material, respectively

$\dfrac{d^2L}{d\Theta^2}$ = specified cam lift acceleration at cam angle Θ

X, Y = transcedental functions giving dimensions of cam and tappet elliptical pressure area

v = auxiliary angle determining the conformity of the tappet face and cam in the vicinity of the elliptical contact pressure area

Fig. 2—The complete contact stress equation can be solved to give the maximum contact stress between a spherical faced tappet and a taper-faced cam. The complete equation takes into consideration the effects of the angle of taper on the cam face and the difference between the specified flat cam lift L and the actual cam lift L_r of a spherical faced follower on a taper-faced cam (Fig. 1b).

To solve the contact stress equation by means of nomography required simplifying the complete equation for easier charting. The complete equation was simplified by rearranging several terms in order to remove the cube root radical from the numerator and also eliminating the terms $(1 + m^2)^{1/2}$, mZ, and $1/(1 + m^2)^{1/2}$ which corrected the equation for the effect of taper on the cam face. Also, L_r was replaced by L. This eliminated a series of nomograms which would have been required to compute L_r. It was known that the effect these terms would have on the final answer would appear beyond the three significant digits considered to be within the range of slide rule accuracy. Since the nomograms for the contact stress equation were intended to have an accuracy equivalent to that of a slide rule, elimination of these terms was considered justifiable. The simplified contact stress equation also can be applied to find the contact stress between a spherical faced tappet and a flat-faced cam (Fig. 1c).

Since maximum contact stress occurs under conditions of static or very low engine speeds, the terms defining the force P compressing the cam and tappet together in the complete and simplified stress equations are for static valve gear loading conditions. In regard to the static loading conditions, it was assumed that the valve gear was rigid, the effects of frictional forces acting on the valve gear were negligible compared to the total valve gear load, the effect of minor angularities between the push rod and tappet center lines was negligible, and that the rocker arm ratio was constant. The force terms in the contact stress equations are further qualified in that they apply only to an overhead valve type of valve gear which uses a single rocker arm to transmit motion from the cam datum to the valve datum.

encountered in selecting the best type of nomogram to fit a given equation and the limits of its equated variables.

The accuracy of a nomogram will not compare with the absolute accuracy of a desk calculator or digital computer. However, in the right application, a nomogram can be a highly useful and desirable tool for the engineer or draftsman.

Stress Equation Divided into Sub-equations for Easy Charting

Charting the cam and tappet contact stress equation (Fig. 2) for nomographic solution required four steps:

(a) Simplification of the complete contact stress equation to make it easier for charting

(b) Breakdown of the simplified equation into sub-equations

(c) Rearrangement of the sub-equations into the general form required for constructing a nomogram

(d) Individual charting of the sub-equations, thus making a series of nomograms necessary for solving the stress equation.

The slight simplification of the complete contact stress equation was done with the knowledge that there would be a negligible effect on the accuracy of the answer.

To make the simplified equation easier to work with, it was necessary to reduce it further by substitution as follows:

$$S_m = \frac{3P}{2\pi XY\left[\dfrac{3P\,(A_1 + A_2)}{8\left(\dfrac{2}{R_f} + \dfrac{1}{R_c}\right)}\right]^{2/3}}$$

$$S_m = \frac{3P}{2\pi XYH^2}$$

where

$$H = \left[\frac{3P\,(A_1 + A_2)}{8\left(\dfrac{2}{R_f} + \dfrac{1}{R_c}\right)}\right]^{1/3}$$

$$X \text{ and } Y \propto \cos v$$

$$\cos v = \frac{\dfrac{1}{R_c}}{\dfrac{2}{R_f} + \dfrac{1}{R_c}}.$$

This reduction involved merely substituting into the simplified contact stress equation appropriate symbols to represent groups of terms associated with various parameters in the equation. These parameters included the cam radius of curvature R_c, the load P compressing the cam and tappet together, and the elastic constants related to the tappet face and cam materials, A_1 and A_2, respectively.

The next step was to write the necessary sub-equations for constructing the required nomograms. Since a series of nomograms was required to chart the contact stress equation, the sub-equations had to be selected and charted in a sequence, using as few nomograms as possible, with the final nomogram solving the contact stress equation in its simplest form.

The substitutions made for the further reduction of the simplified stress equation also provided the basis for selecting and charting the first three sub-equations. The value for at least one of these three parameters (R_c, P, and $A_1 + A_2$) was required in all the remaining sub-equations. Therefore, sub-equations for R_c, P, and $A_1 + A_2$ were written first. The remaining sub-equations were written as necessary to continue the reduction of the simplified equation until it could be charted in the final nomogram.

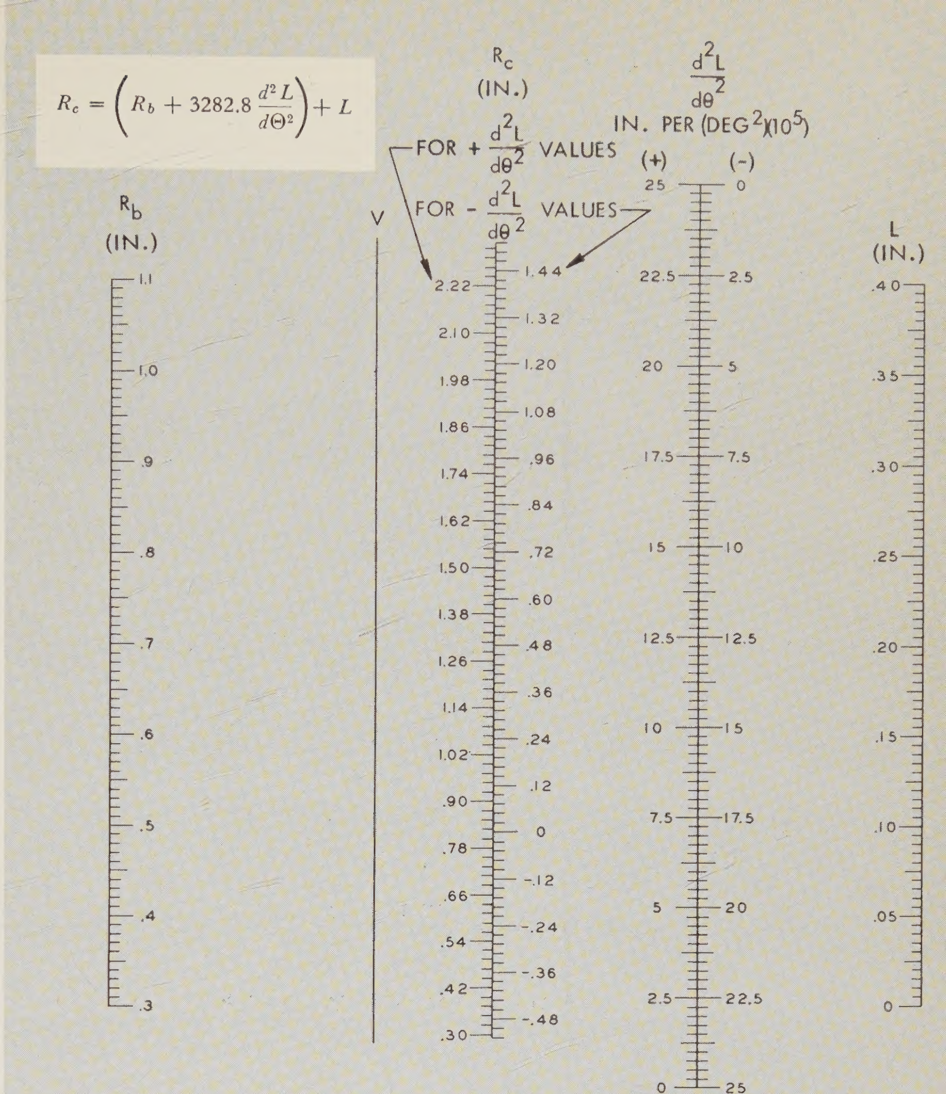
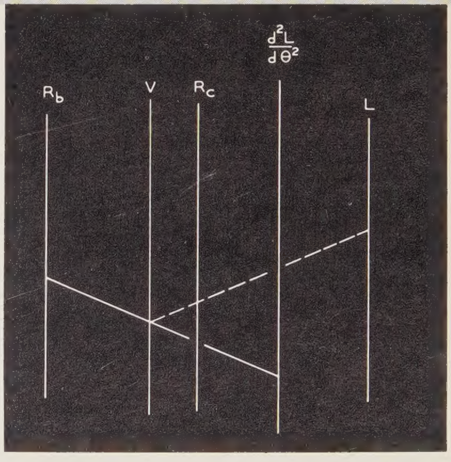


Fig. 3—The sub-equation for R_c, the cam radius of curvature, was relatively simple in that it involved only the summation of three terms. The lift acceleration term $d^2L/d\Theta^2$, however, could be either positive or negative. Consequently, the summation was algebraic which slightly complicated charting of the nomogram. Since the equation was a summation type, a nomogram with equally divided vertical scales was used to solve the equation. The constant *3282.8* was incorporated into the scale for the $d^2L/d\Theta^2$ variable. To accommodate both positive and negative values of lift acceleration, the $d^2L/d\Theta^2$ variable and the final answer R_c required dual scales. The equally divided answer scale was developed especially for this nomogram. The diagram at the lower right hand corner indicates the procedure used to find R_c. The solid line is the intermediate solution line which is drawn first using known data. The dashed line is the final solution.



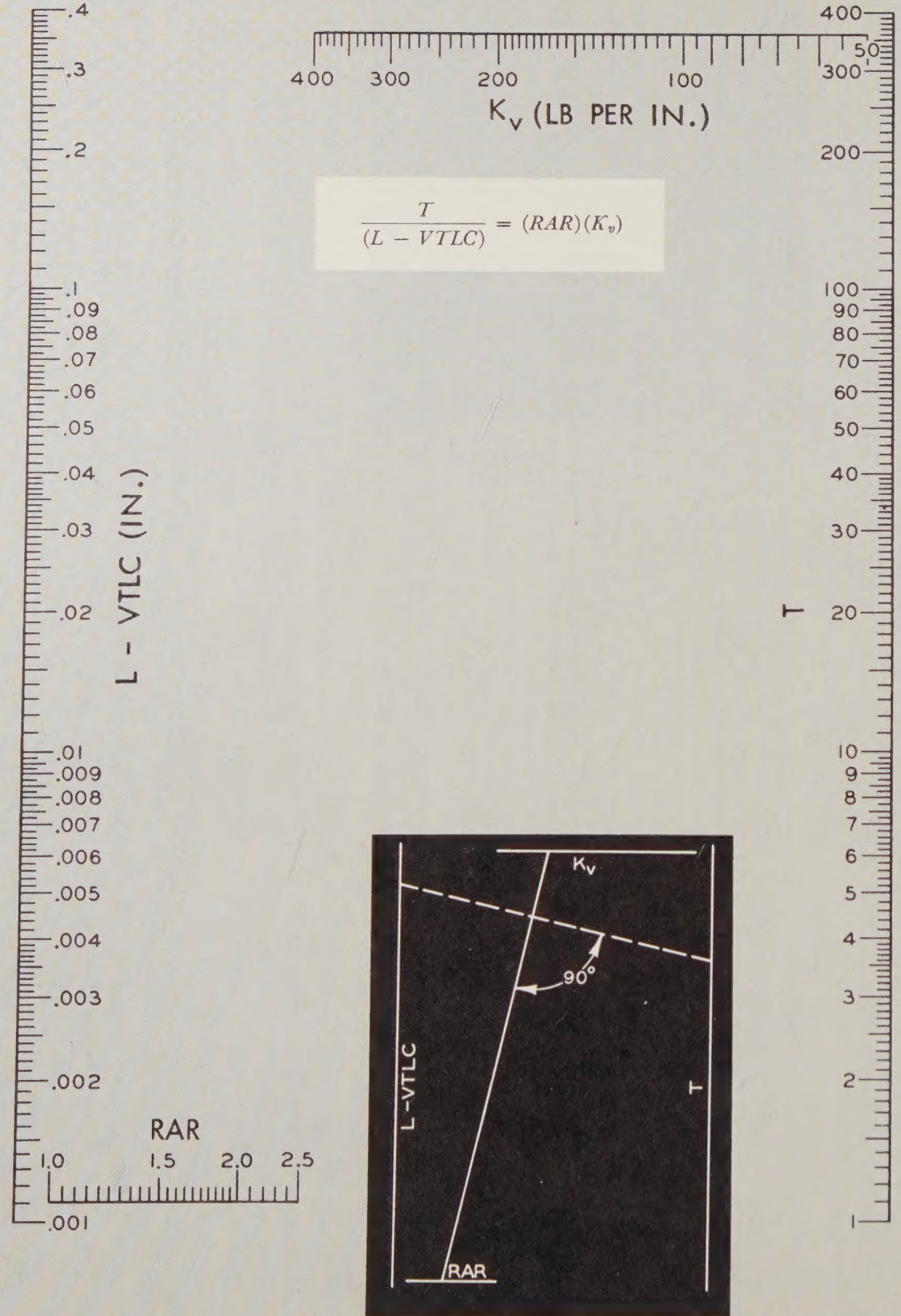
Seven Nomograms Used to Solve Stress Equation

The first sub-equation charted was the equation for the cam radius of curvature R_c. The sub-equation was written as

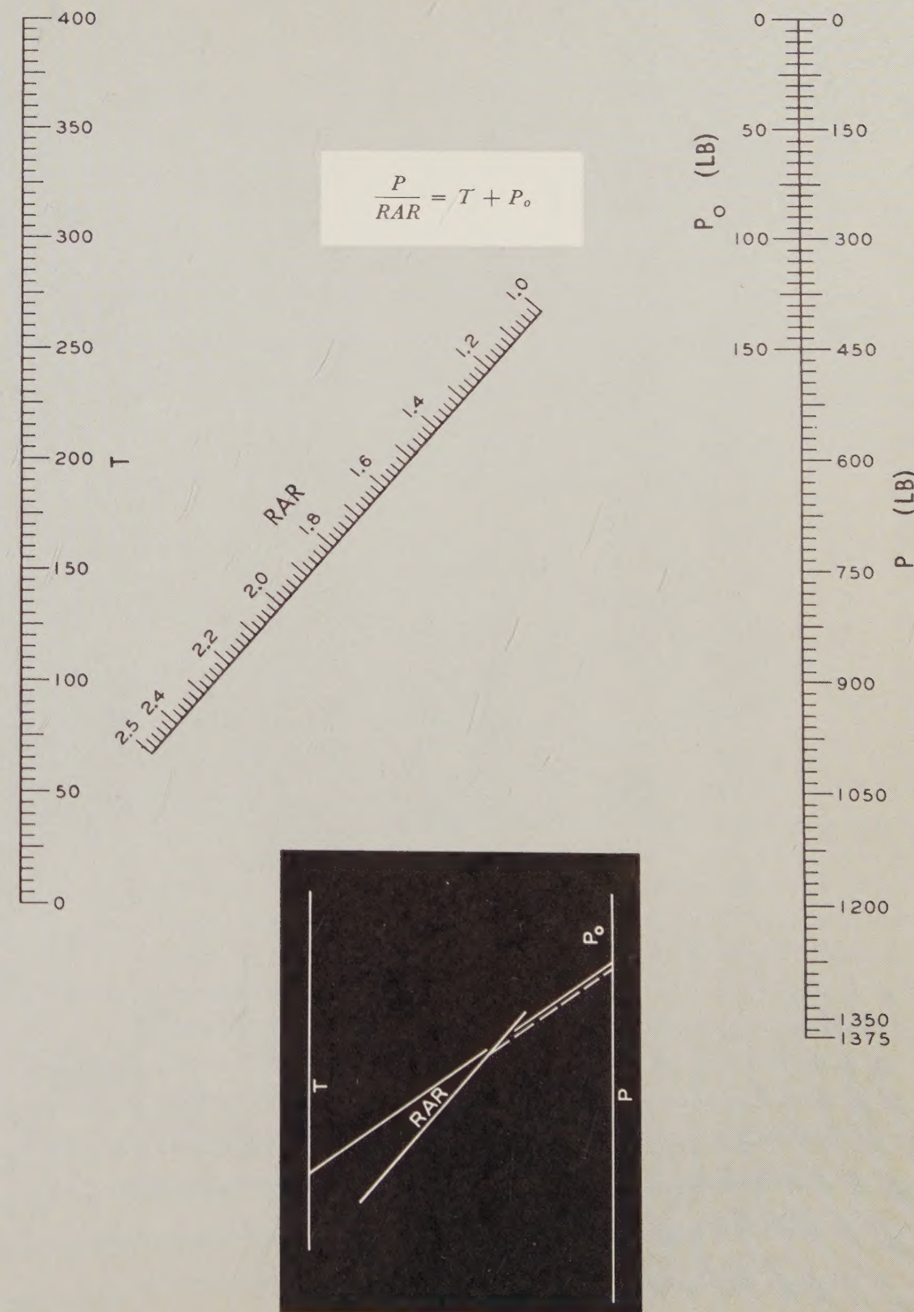
$$R_c = L + R_b + 3282.8\left(\frac{d^2L}{d\Theta^2}\right).$$

For charting purposes, the sub-equation was rewritten as follows:

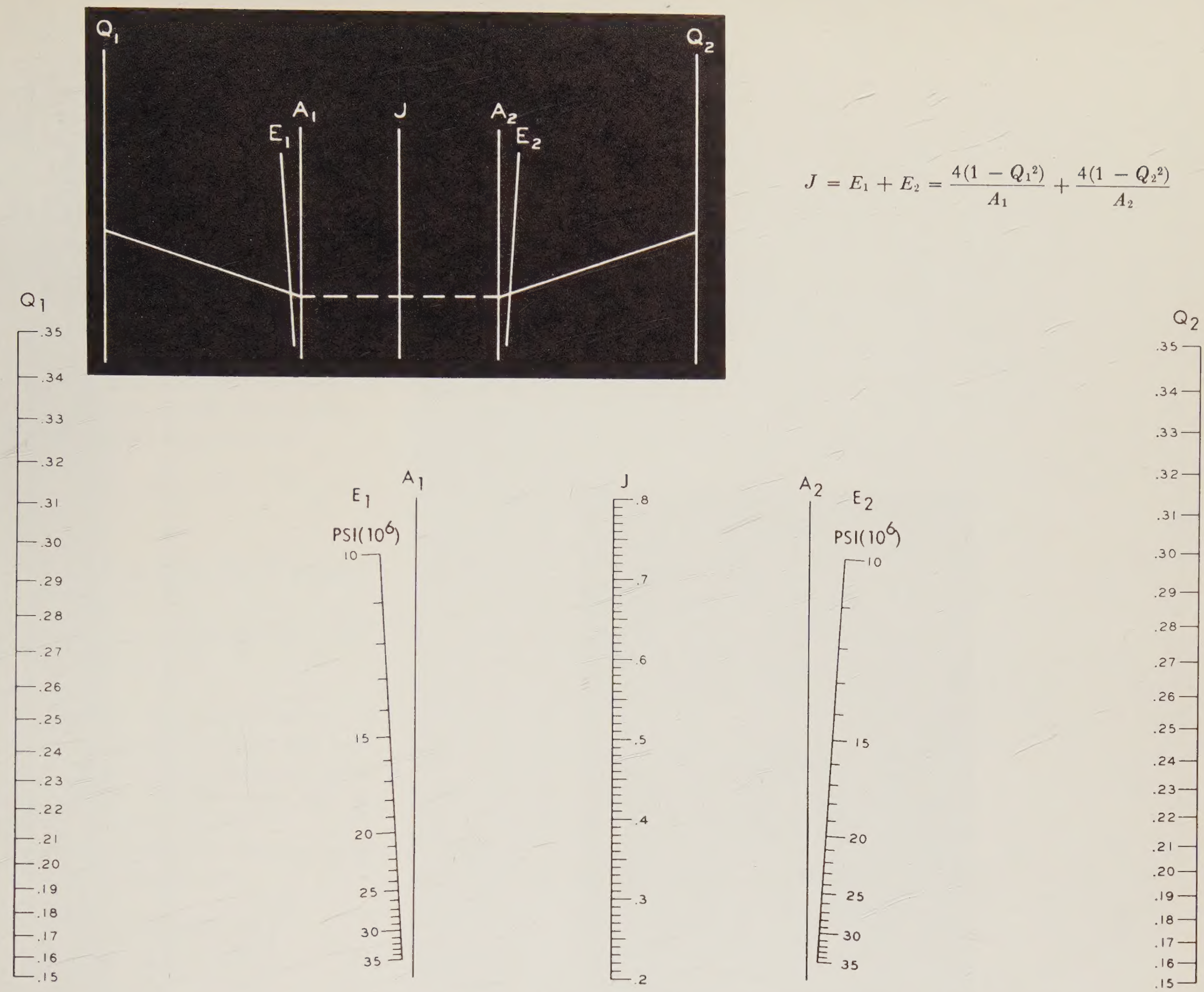
$$R_c = \left(R_b + 3282.8\,\frac{d^2L}{d\Theta^2}\right) + L.$$



$$\frac{T}{(L - VTLC)} = (RAR)(K_v)$$



$$\frac{P}{RAR} = T + P_o$$



$$J = E_1 + E_2 = \frac{4(1 - Q_1^2)}{A_1} + \frac{4(1 - Q_2^2)}{A_2}$$

Fig. 5—The sub-equation for the J factor involved both addition and division. To have scale compatibility, therefore, a Z-type nomogram was used. In this case, two Z charts were used back to back to perform the division part of the equation. The two vertical scales for A_1 and A_2 were then used in a simple vertical summation chart to obtain the answer for J. Scale divisions were omitted for A_1 and A_2 since they contributed nothing to the overall nomogram. The E scales were not equally divided on the nomogram since they were solution scales for the Z charts and had to be developed accordingly. Charting the Q variables required the use of scales representing a function of the variable $-f(Q)$ scales. The $f(Q)$ scales were equally divided when charted. When the values for the Q variables were superimposed on the $f(Q)$ scales, however, the resulting Q scales were divided unequally, as indicated. The diagram at the top indicates the procedure used to solve the sub-equation for the J factor. The solid lines are the intermediate solution and the dashed line is the final solution.

Fig. 4—The sub-equation for the static valve gear load P compressing the cam and tappet together contained six variables which were too many for one nomogram. The sub-equation, therefore, was rewritten into the two equations shown here, each of which contained four variables. To chart the equation at the left, which involved both multiplication and division, a nomogram having logarithmic scales was used. This nomogram could have been constructed with either vertical scales and parallel solution lines or with both vertical and horizontal scales and perpendicular solution lines. The latter type of nomogram was selected to avoid crowding of the scales.

To chart the equation at the right required a nomogram which would allow both summation and division of variables. The standard nomogram requires equally divided scales for summation of variables and logarithmic scales for multiplication or division of variables. Since logarithmic and equally divided scales would have been incompatible, a special version of the Z-type nomogram was used. The Z-type nomogram can be used for division and, in special cases, for multiplication with equally divided scales. The Z-type nomogram used here required parallel solution lines. The diagram shown with each nomogram illustrates the procedure used to solve the equation. The solid line is the intermediate solution and the dashed line is the final solution.

Since the equation for R_c was a summation type, a nomogram with equally divided vertical scales was used to solve the equation (Fig. 3).

The second sub-equation charted was the equation for the static valve gear load P compressing the cam and tappet together. The complete sub-equation was

$$P = (RAR)^2 (K_v) (L - VTLC) + (P_o) (RAR).$$

This equation contained six variables which, in this case, were too many for one nomogram. The equation, therefore, was rewritten into the following two separate equations, each containing four variables:

$$\frac{P}{RAR} = (RAR)(K_v) (L - VTLC) + P_o$$

Let

$$T = (RAR) (K_v) (L - VTLC).$$

Therefore,

$$\frac{T}{(L - VTLC)} = (RAR) (K_v) \qquad \text{(a)}$$

$$\frac{P}{RAR} = T + P_o. \qquad \text{(b)}$$

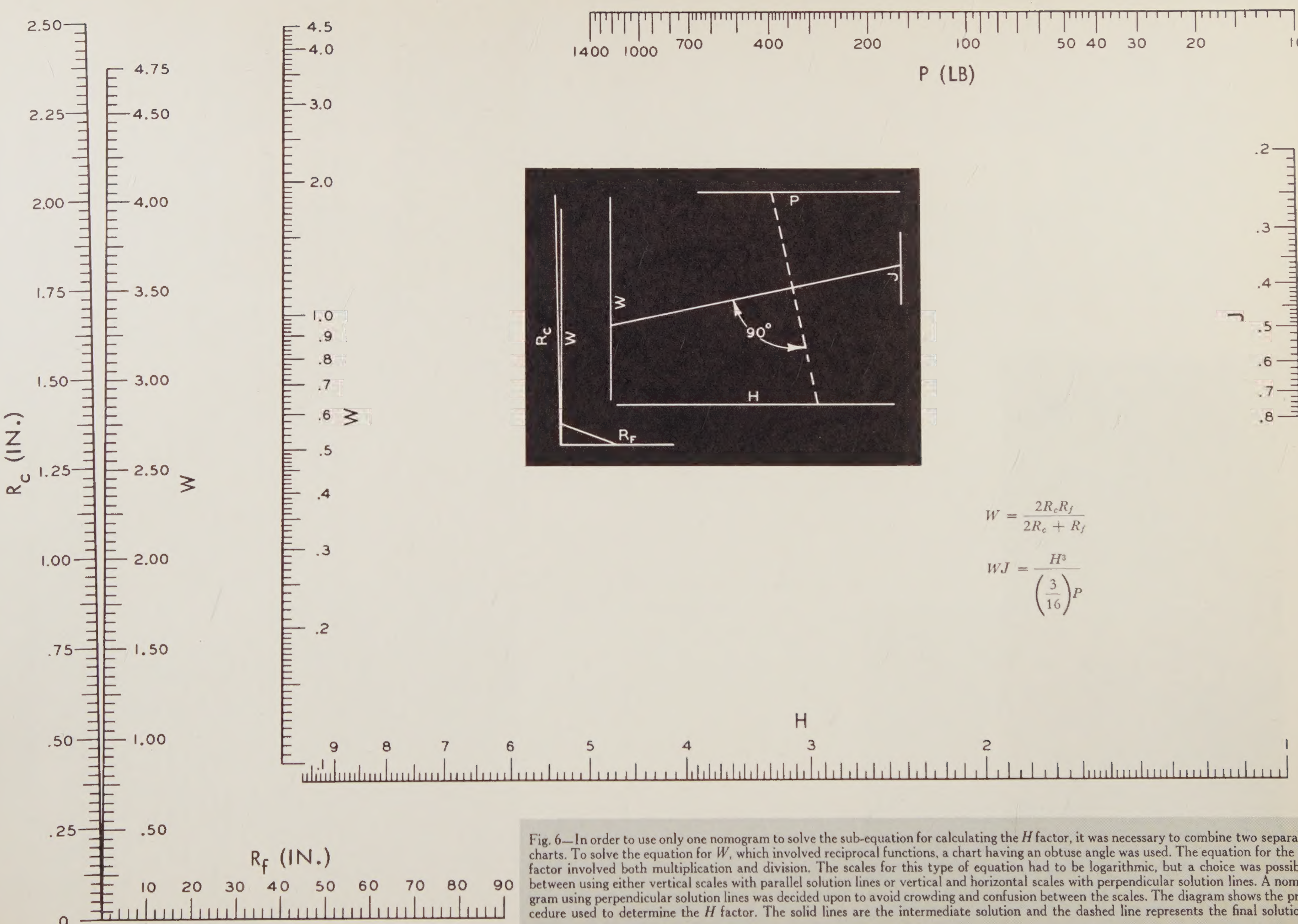


Fig. 6—In order to use only one nomogram to solve the sub-equation for calculating the *H* factor, it was necessary to combine two separate charts. To solve the equation for *W*, which involved reciprocal functions, a chart having an obtuse angle was used. The equation for the *H* factor involved both multiplication and division. The scales for this type of equation had to be logarithmic, but a choice was possible between using either vertical scales with parallel solution lines or vertical and horizontal scales with perpendicular solution lines. A nomogram using perpendicular solution lines was decided upon to avoid crowding and confusion between the scales. The diagram shows the procedure used to determine the *H* factor. The solid lines are the intermediate solution and the dashed line represents the final solution.

The complete sub-equation for the static valve gear load P was rewritten to take best advantage of apparent scale size requirements and scale compatibility. The rewritten equation (a) was of a type involving both multiplication and division. To solve this equation a nomogram having logarithmic scales was used (Fig. 4-left). To solve equation (b) a Z-type nomogram having parallel solution lines was used to allow both summation and division of variables (Fig. 4-right).

The next sub-equation charted was for the J factor relating the material constants in the contact stress equation. This sub-equation was written as

$$J = A_1 + A_2 = \frac{4\,(1 - Q_1^2)}{E_1} + \frac{4\,(1 - Q_2^2)}{E_2}.$$

For charting purposes the sub-equation was rewritten as

$$J = E_1 + E_2 = \frac{4\,(1 - Q_1^2)}{A_1} + \frac{4\,(1 - Q_2^2)}{A_2}.$$

The rewritten sub-equation for the J factor involved both addition and division. To have scale compatibility, therefore, a Z-type nomogram was used to solve the equation (Fig. 5).

The sub-equation for charting the H factor was written next as:

$$H = \left[\frac{3P\,(A_1 + A_2)}{8\left(\dfrac{2}{R_f} + \dfrac{1}{R_c}\right)}\right]^{1/3}$$

This equation was then rewritten as follows for easier charting:

$$H = \left[\frac{3R_c\,R_f\,PJ}{8\,(2R_c + R_f)}\right]^{1/3}$$

$$H^3 = \frac{R_c\,R_f}{8\,(2R_c + R_f)}\,(3PJ)$$

$$H^3 = \left(\frac{2R_c\,R_f}{2R_c + R_f}\right)\left(\frac{3}{16}\,P\right)(J)$$

Let

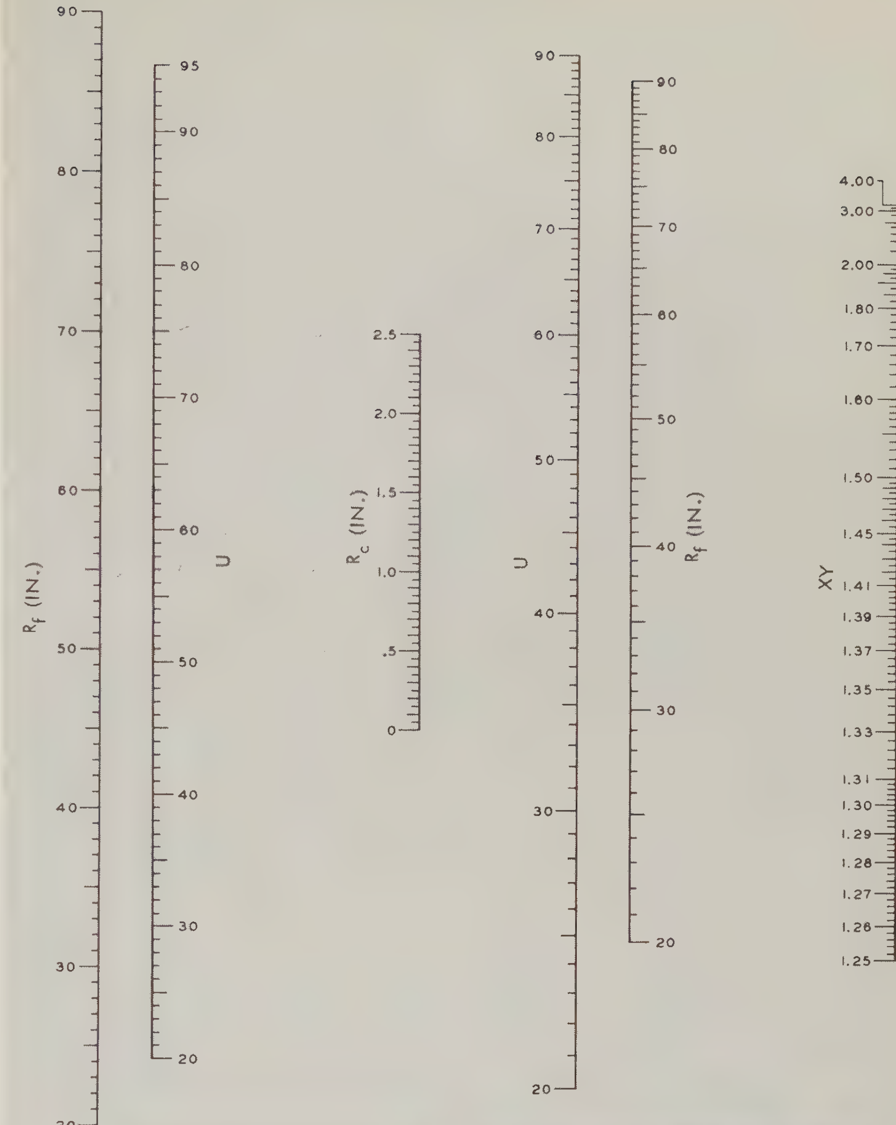
$$W = \frac{2\,R_c\,R_f}{2\,R_c + R_f}$$

Then

$$H^3 = W\left(\frac{3}{16}\,P\right)(J)$$

Or,

$$(W)\,(J) = \frac{H^3}{\left(\dfrac{3}{16}\,P\right)}.$$



$$2R_c + R_f = U$$

$$(U)(XY) = R_f$$

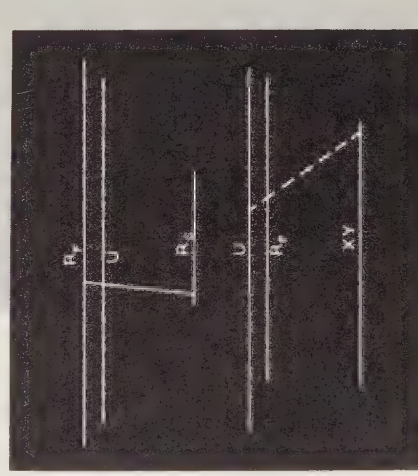


Fig. 7—Shown here is the nomogram used to solve the sub-equation for evaluating the XY factor. To solve this sub-equation required a nomogram having both equally divided and logarithmic vertical scales. The diagram at the lower right illustrates the procedure followed to determine XY. The solid line is the intermediate solution determined from known data. The dashed line represents the final solution.

The sub-equation for calculating the H factor was derived from the quantity in the denominator of the complete simplified contact stress equation (Fig. 2) raised to the two-thirds power and the subsequent further reduction of the equation shown previously. The simplified sub-equation for the H factor was charted by combining two separate charts into a single nomogram (Fig. 6).

Before the final solution could be obtained, one more sub-equation was needed. This was the sub-equation for evaluating the XY factor and was written as

$$X \text{ and } Y \propto \cos v$$

$$\cos v = \frac{\dfrac{1}{R_c}}{\dfrac{2}{R_f} + \dfrac{1}{R_c}}$$

Fig. 8—The simplified equation for maximum contact stress was rewritten into an equation having multiplication and division of four variables. This type of equation required a nomogram having logarithmic scales using either parallel or perpendicular solution lines, depending on the layout of the nomogram. To avoid crowding and confusion of the scales, a nomogram using perpendicular solution lines was favored. The constant 2π in the equation was combined with the XY variable and the constant *3* was combined with the P variable. The square of the H variable was incorporated into the H scale. The diagram illustrates the procedure used to obtain a value for S_m. The solid line represents the intermediate solution and the dashed line represents the final solution.

For charting purposes, the sub-equation was written as

$$\cos v = \frac{R_f}{2R_c + R_f}$$

$$R_f = (2R_c + R_f)\ (\cos v)$$

$$R_f = (2R_c + R_f)\ (XY)$$

First attempt at charting:

$$2R_c + R_f = \frac{R_f}{XY}$$

Second attempt at charting:

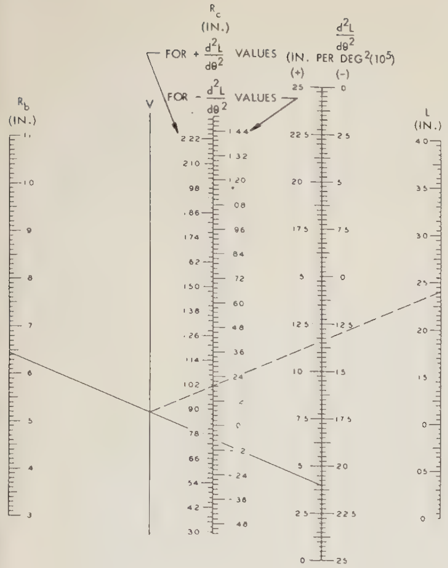
$$2R_c + R_f = U$$
$$(U)\ (XY) = R_f.$$

Values of X and Y were obtained directly from curves of $\cos v$ versus X and Y, which were developed through the evaluation of complete elliptical integrals of the first and second order. To make a nomographic solution of this equation, therefore, data from these curves were used to develop a scale for the XY variable. Since the equation could be written into the type form having both addition and division, the first attempt at charting the equation was to use a special version of the Z-type nomogram. The nomogram was constructed using the actual scale for $\cos v$. The products of X and Y for corresponding values of $\cos v$ then were superimposed on the $\cos v$ scale. However, the first attempt at charting the sub-equation failed since the Z-type nomogram caused the scale for XY to become compressed over a critical range, thus impairing accuracy. A change to an alternate type of nomogram (Fig. 7) having vertical scales, both equally divided and logarithmic, corrected the situation and provided an adequate solution to the sub-equation.

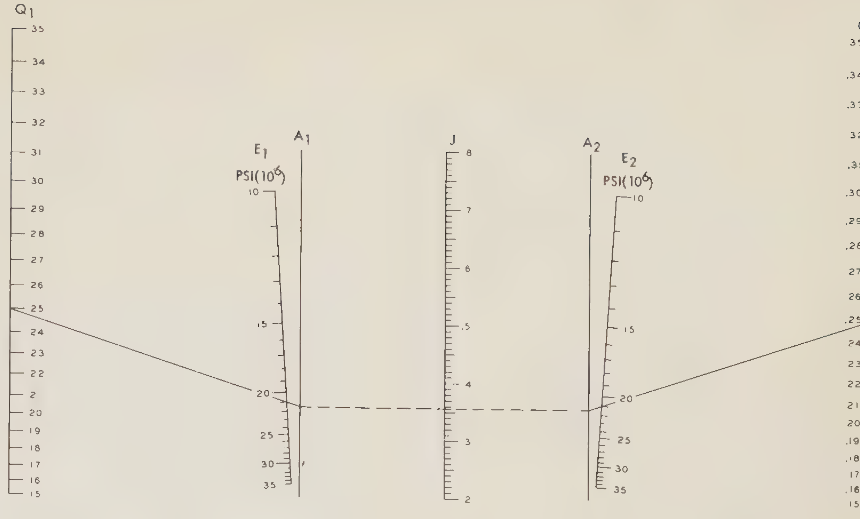
The final nomogram developed solved the simplified equation for the maximum contact stress. The simplified equation was

$$S_m = \frac{3P}{2\pi\ (XY)\ (H^2)}$$

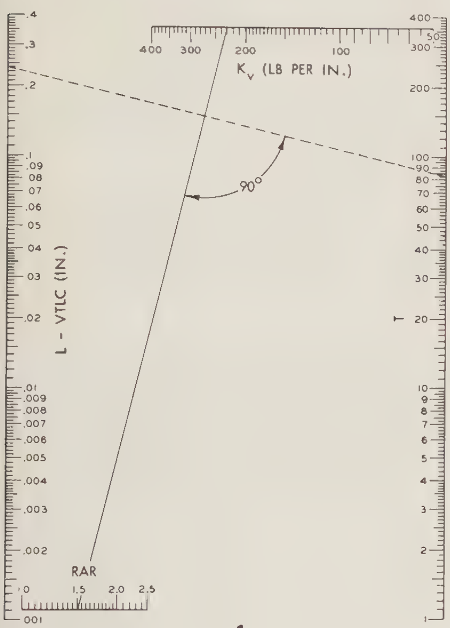
Fig. 9—The seven nomograms developed to solve the cam and tappet contact stress equation includes those having single, perpendicular, and parallel solution lines. Each nomogram uses a solid line for the intermediate solutions(s) and a dashed line for the final solution. The solution is achieved by solving each nomogram in sequence, starting with the first, until the final answer is obtained. In each nomogram the intermediate solution(s) is found first using given data and/or the answer from the preceding nomogram(s). The final solution in each nomogram is found using the intermediate solution and a single, perpendicular, or parallel solution line. For example, the sample problem given to illustrate the use of the nomograms in finding the maximum contact stress at $-16°$ of cam rotation included values for R_b, $d^2L/d\Theta^2$, and L in the given data. Nomogram (a) is the first nomogram used in solving the problem. A straight line is drawn between the respective values of R_b and $d^2L/d\Theta^2$. This line intersects the V scale and provides the intermediate solution. A straight line then is drawn from the intersection point on the V scale to the respective value on the L scale. This line intersects the R_c scale, giving the final solution for R_c. Nomogram (b) is used next. The values of RAR, K_v, and $L-VTLC$ were given in the sample problem. A straight line is drawn between the respective values on the RAR and K_v scales. This line provides the intermediate solution. The final solution line is drawn perpendicular to the intermediate solution line from the $L-VTLC$ scale to the T scale, giving the value for T. Nomogram (c) is solved in the same manner as nomogram (b), except parallel solution lines are used. The values for RAR and P_o were given in the problem, while the value for T was obtained from nomogram (b). Therefore, a straight line drawn between the respective values on the T and P_o scales provides the intermediate solution, while the final solution for P is obtained by drawing a line parallel to the intermediate solution line from the respective value on the RAR scale to the P scale. The remaining nomograms are solved in a similar manner until a final value of 146,000 psi for S_m is obtained in nomogram (g).


R_b (IN.)
V
R_c (IN.)
FOR + $\frac{d^2L}{d\theta^2}$ VALUES
FOR − $\frac{d^2L}{d\theta^2}$ VALUES
$\frac{d^2L}{d\theta^2}$ (IN. PER DEG^2)(10^5)
(+)
(−)
L (IN.)

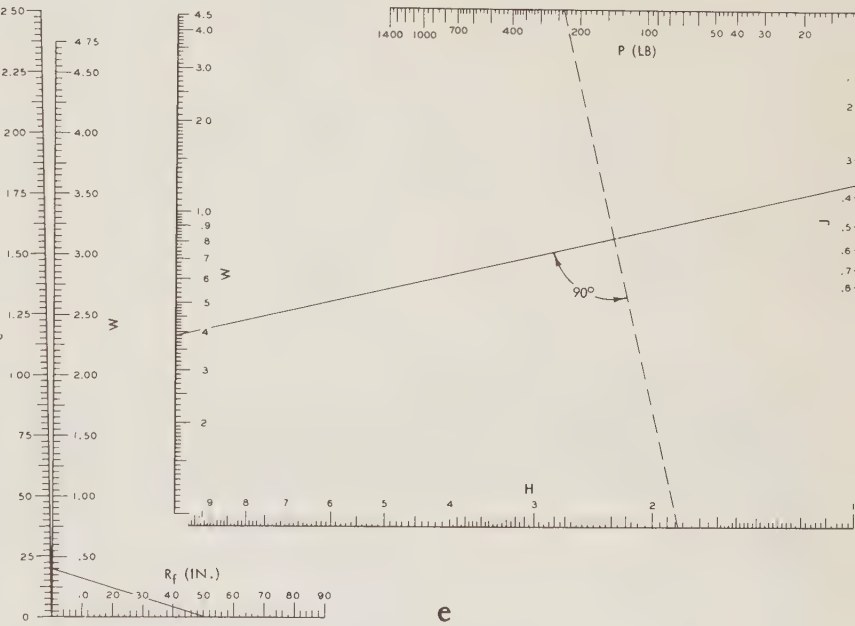
a


Q_1
E_1 PSI(10^6)
A_1
J
A_2
E_2 PSI(10^6)

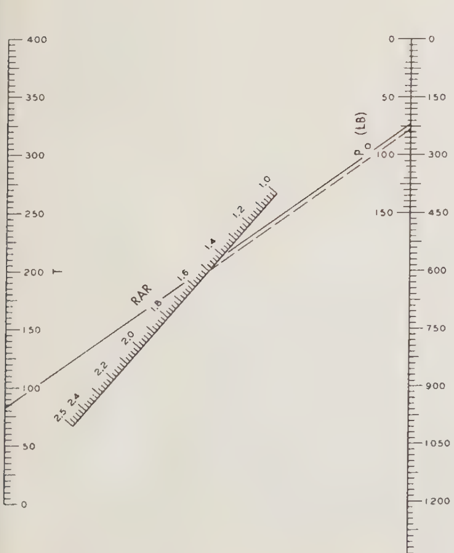
d


L - VTLC (IN.)
K_v (LB PER IN.)
T
RAR
90°

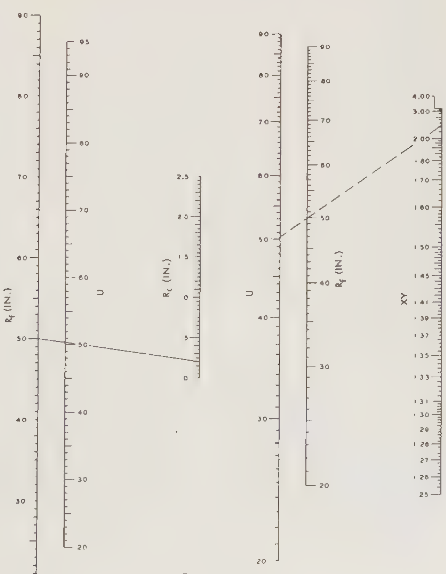
b


R_c (IN.)
W
R_f (IN.)
P (LB)
J
H
90°

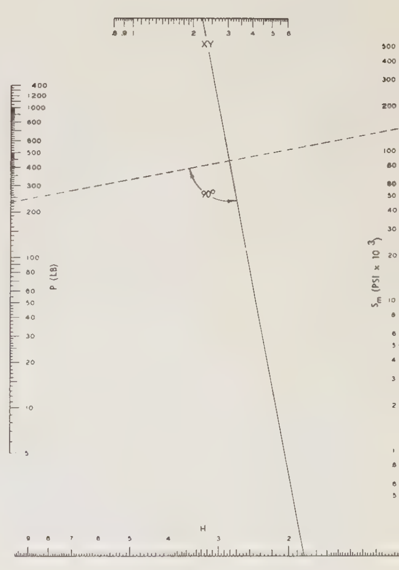
e


T
RAR
P_o (LB)
P (LB)

c


R_f (IN.)
U
R_c (IN.)
XY

f


XY
P (LB)
H
90°

g

This equation was rewritten for charting purposes as

$$(2\pi XY)\ (H^2) = \frac{3P}{S_m}.$$

The nomogram developed to solve this equation used perpendicular solution lines to avoid crowding and confusion of scales (Fig. 8).

Sample Problem Illustrates Use of Nomograms

To illustrate the use of the seven nomograms developed to solve the contact stress equation, consider the following problem for a spherical faced tappet and taper-faced cam combination.

Given data:

R_f = 50.0 in.
R_b = 0.64560 in.
RAR = 1.50 to 1
K_v = 230.8 lb per in.
P_o = 74 lb
$VTLC$ = 0 in. (hydraulic valve lifter used)
Q_1, Q_2 = 0.25
E_1, E_2 = (21) (10^6) psi
Θ = $-16°$
L = 0.238850 in.
$\frac{d^2L}{d\Theta^2}$ = -0.00021 in. per degree squared

Find:

The maximum contact stress at $-16°$ of cam rotation (Fig. 9).

The maximum contact stress obtained by using the nomograms is 146,000 psi. By comparison, the answer to the same problem as solved on a digital computer using the complete stress equation was 145,380 psi. The per cent error between the nomographic solution and the computer solution is

$$\frac{145{,}380 - 146{,}000}{145{,}380}\ (100)$$

$$= -0.43 \sim -0.5 \text{ per cent.}$$

The error of minus one-half of one per cent includes the error deliberately introduced when the original equation was simplified to make it applicable to solution by nomography. This amount of error compares very favorably with slide rule accuracy.

Summary

Nomograms can be used to solve most equations encountered by an engineer. The accuracy obtained with a nomogram will depend on the type of equation charted, the complexity of the equation, the size of the nomogram and its respective scale divisions, the magnitude and limits of the equated variables, and the dexterity of the individual using the nomogram.

In general, the accuracy of a nomogram can be considered equivalent to or better than that of a slide rule. For example, the sample problem used to illustrate the use of the seven nomograms developed to solve the cam and tappet contact stress equation showed that the nomograms gave an answer having an accuracy of minus one-half of one per cent on an answer in the order of 150,000 psi.

Pitfalls do exist in nomography, especially in the selection of the type of nomogram to use when charting a given equation. With a little foresight and planning, however, a given equation can be charted without too much difficulty.

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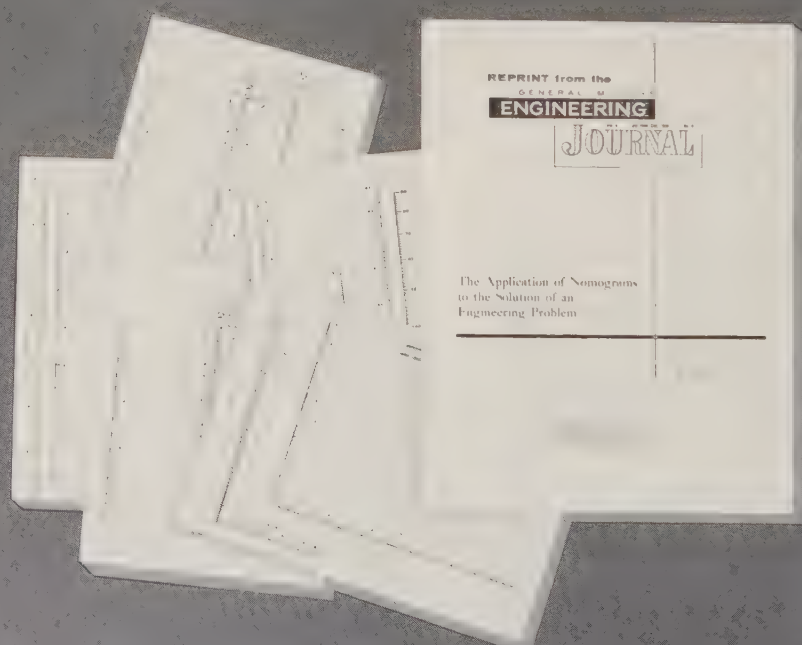
REPRINT AVAILABLE

Reprints of the paper "The Application of Nomograms to the Solution of an Engineering Problem" and sets of full-size nomograms discussed in the paper (Figs. 3 through 8) are available for classroom use. Educators interested in receiving either a reprint of the paper or a reprint of the paper together with a set of the nomograms may write to Educational Relations Section, Public Relations Staff, GM Technical Center, P.O. Box 177, North End Station, Detroit 2, Michigan.



Design of a Final Calibration Stand for a Pendulous Integrating Gyroscope

By GEORGE E. SONNTAG
and JOHN L. PERAMPLE
AC Spark Plug Division

One of the distinctive characteristics of an effective inertial guidance system is the extreme accuracy of indicated position which it provides for the guided vehicle. Extreme accuracy, in turn, is a characteristic of the components of such a system. Problems of accuracy and precision are especially significant in the gyroscopes which are the fundamental position sensors of the inertial guidance system. At the Milwaukee Plant of AC Spark Plug Division, where inertial guidance systems are built for several types of missiles, it is necessary to develop a variety of special calibration and testing fixtures to insure that a device or system meets the stringent requirements of accuracy. A recent example was the design of an improved final calibration stand for a pendulous gyroscope used in an integrating accelerometer for a ballistic missile. The requirement was to apply a known acceleration to the gyroscope. Instead of the centrifuge method, this calibration stand tilted the gyroscope so that the gravity vector caused an acceleration input. Advantages were improved servo loop performance, easier control of the test, and more accurate calibration.

The test fixture must be better than the device being tested

INERTIAL guidance systems provide phenomenal accuracies to guided vehicles. Although these systems presently are employed in ballistic and air breathing missiles, they easily could guide aircraft with passengers or freight from New York to Los Angeles without pilot corrections.

A guidance system for a ballistic missile provides control only during the initial powered phases of the missile's flight. The guidance system aims the missile during acceleration and then cuts off the power when the velocity necessary to hit the target has been reached. As an example of the accuracies required in missile guidance, an error of one ft per sec out of the 24,000-ft per sec velocity of the missile at the power cutoff point will result in a one-mile miss at the target.

All guidance systems consist of two principal elements:

(a) A sensor which measures missile motion and has an accuracy in the order of one part in 100,000

(b) A computer which calculates the missile velocity from the sensor information, compares this velocity to the velocity required to reach the target, and gives steering orders and power cutoff orders to the missile autopilot.

Present guidance systems differ primarily in the type of sensor used. Some sensors are radio controlled and include ground based equipment. The all-inertial system, on the other hand, is contained within the missile. It consists essentially of a gyro stabilizer and platform which carries delicate integrating accelerometers. The accelerometers measure the missile acceleration and convey this information to a digital computer carried within the vehicle (Fig. 1). At the Milwaukee Plant of AC Spark Plug Division, where inertial guidance systems are built, it is necessary to test and calibrate with extreme accuracy the components making up these systems. An example is the testing of a pendulous gyroscope used in an integrating accelerometer. To improve existing methods for testing accelerometers, AC engineers recently developed a new test fixture.

Precession Rate Proportional to Acceleration

An integrating accelerometer displays its output as a function of output shaft position. This shaft position is usually measured by a potentiometer coupled to the output shaft (Fig. 2).

The accelerometer contains a complete independent servo loop which integrates the input accelerations. In this loop the instantaneous rate of the output shaft is a measure of the instantaneous acceleration. The linear acceleration is proportional to angular velocity. Linear velocity, therefore, is proportional to the angular position.

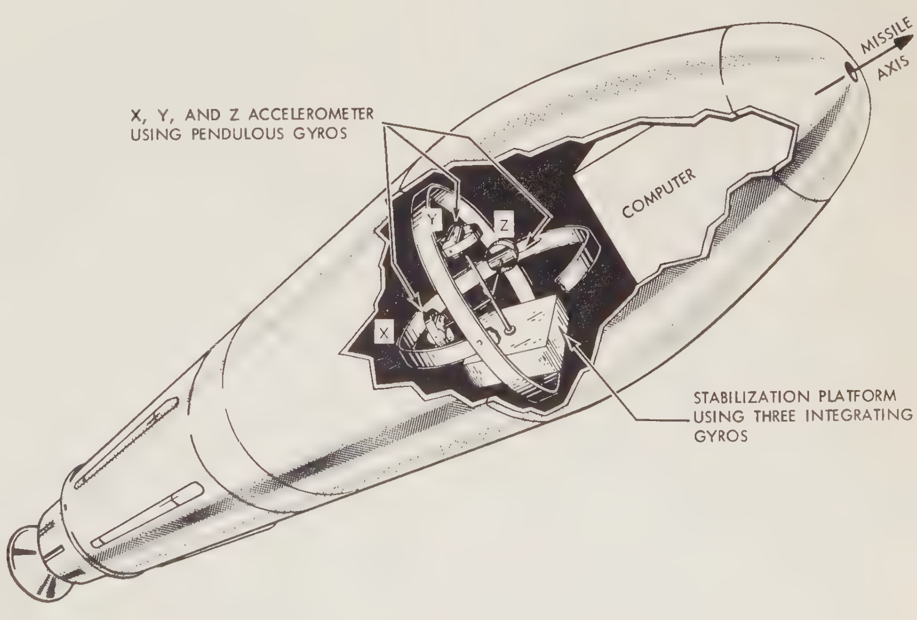


Fig. 1—Shown here are the principal elements of the inertial guidance system in a missile vehicle. The accelerometers transmit information to the computer which gives control orders to the missile autopilot.

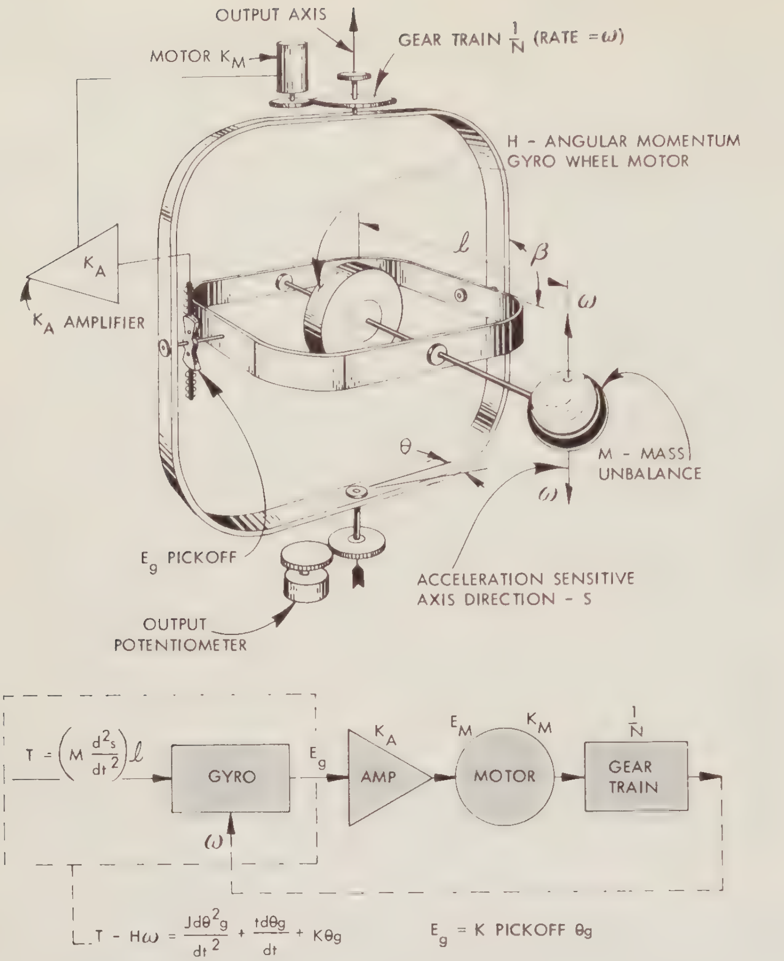


Fig. 2—This drawing illustrates the application of the pendulous gyroscope in accelerometers and the servo loop which integrates the input accelerations of the missile. The instantaneous rate of the output shaft is a measure of the instantaneous acceleration. The output shaft rate is measured by the output potentiometer.

The gyroscope is constructed so that a pendulous mass produces a torque about the output axis of the gyroscope when it is accelerated in the direction of the gyroscope *input axis*. This torque can be counteracted by rotating the gyroscope about its input axis. This is accomplished by a servomotor, K_M (Fig. 2).

The basic equation for equilibrium of this system is:

$$Mla = H\omega$$

where

Ml = (mass) (moment arm) = pendulosity

a = acceleration in direction of gyroscope input axis

H = gyroscope wheel momentum

ω = angular rate of rotation about the input axis.

Since M, l, and H are constants, the rate of rotation, ω, is proportional to input acceleration a, and the instantaneous velocity, in the direction of the acceleration, is proportional to the angle through which the gyroscope is rotated about the input axis.

Therefore:

$$\text{Velocity about input axis} = \int a\, dt$$

$$\text{Angular rotation } \Theta = \int \omega\, dt.$$

Device Based on Tilting Principle

To test the gyroscopes it is necessary to apply known accelerations. This is most easily done either by centrifuging or by tilting the gyroscope in such a manner that the gravity vector causes an input. Tilting has a maximum input value of 1g, while centrifuging is limited only by the capacity of the centrifuge mechanism. However, experience indicates a preference for the tilt method for production tests, even though an accelerometer must detect higher acceleration in a missile. Some of the reasons for favoring the tilt method are:

(a) The input is steadier, more repeatable, and more easily controlled

(b) Most of the gyroscope output errors are constant or sinusoidal functions with constant coefficients. They appear more significant at inputs up to 1g and thus can be more easily analyzed

(c) Being a null operating device, the output is proportional to the input at all levels except for errors in (b) above. Saturation types of nonlinearity do not exist.

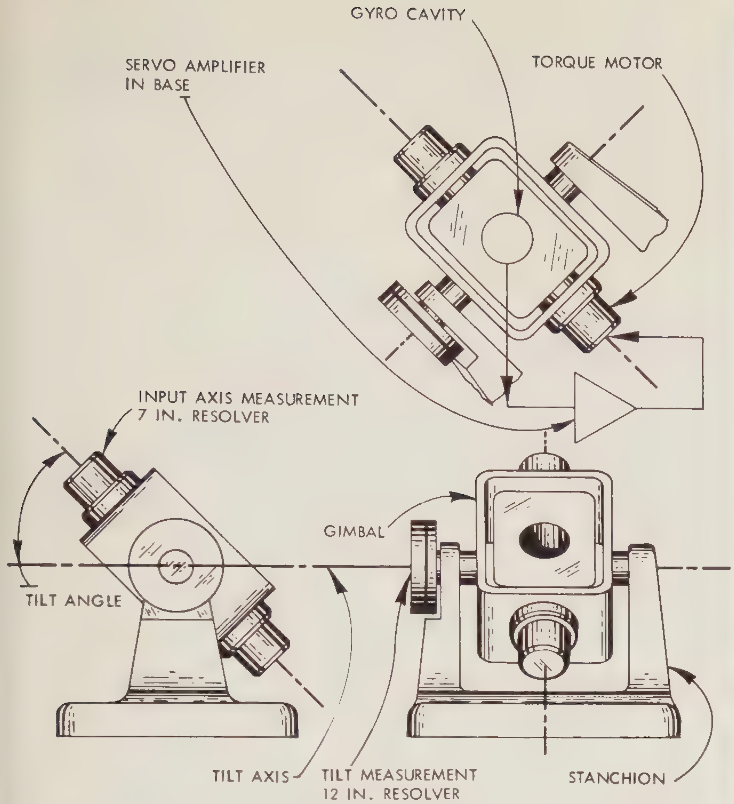
The design of the test stand, therefore, incorporates a gimbal which is free to rotate on stanchion supports. The tilt, or acceleration input, is applied to this gimbal (Fig. 3). A gyroscope to be tested is placed in a cavity of the input axis shaft of the test stand. The shaft is driven by a d-c torque motor at one end and the shaft position, or rate, is measured by a 7-in. diameter resolver at the other end.

One of the design problems of the stand was how to achieve the extreme accuracies required in gyroscope testing. It had been determined that the fixture should not introduce more than 1 microradian per second error in finding the angular rate of rotation of the input axis ω. This requirement implied that the following operational accuracies of the fixture would be necessary:

- Tilt axis measurement to be accurate to 0.5 seconds and repeatable to 0.25 seconds
- Angular rate measurement to be accurate to 1.2 angular seconds per revolution
- Time interval per revolution to be accurate to 10^{-5} seconds
- The servo loop to maintain the pendulum of the gyroscope within 1 second of perpendicularity to the input axis.

Angular Measurement Largest Obstacle

An important phase of the design, therefore, was a study of angular measurement. Mechanical techniques proved basically inadequate, and the search was concentrated on optical and electronic



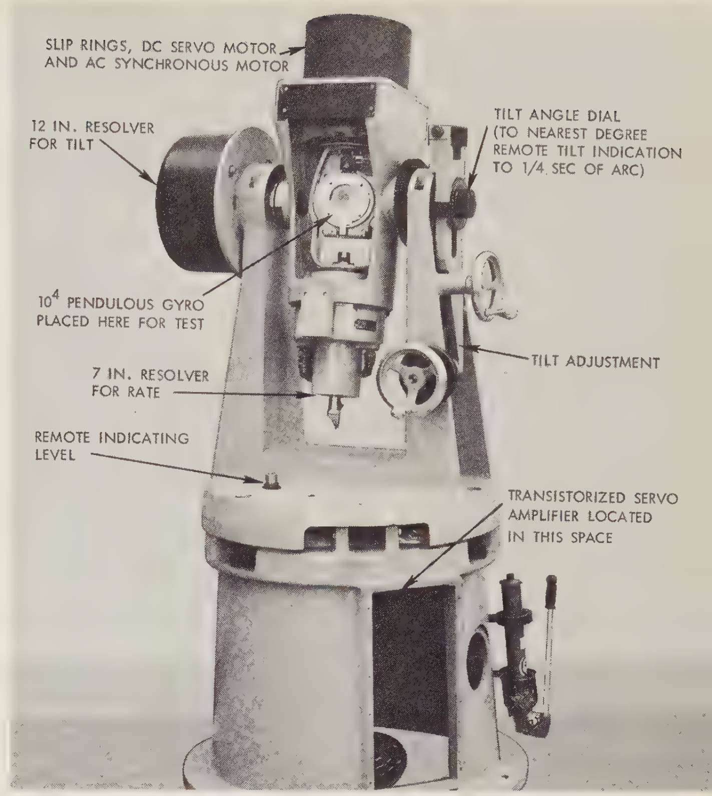


Fig. 3—To test gyroscopes in the new test stand, it was necessary to apply known accelerations. By tilting the gyroscope, the gravity vector causes an acceleration input. The principal components for tilting and measurement are identified in the drawing (left). The gyroscope is placed in a cavity of the input axis shaft which is supported in a gimbal. The shaft is rotated by a direct-connected torque motor. The gimbal can be tilted and the angle measured. The actual test stand is shown at the right. Extreme accuracies in machining and assembling were required in the fabrication of the stand to reduce friction and friction variations and to minimize the effect of any test stand errors on calibration.

techniques that utilized frequency differential, phase shift, capacitance change, light interferometry, magnetic tape, and spectroscopy. Many sources were consulted on these techniques, and included experts in these fields within AC Spark Plug Division and at the GM Research Laboratories.

The method chosen for both tilt and rate measurement in the test stand was the use of an electrical resolver with single turn windings etched in two glass plates to form the stator and rotor (Fig. 4). The extreme accuracy possible with these two resolvers was the result of the precise dividing of the windings on the glass and the averaging effect of multiple series windings. This averaging effect also made the resolvers insensitive to small mounting eccentricities.

A 12-in. diameter, 360-pole resolver was chosen for the tilt measurement device and a 7-in. diameter, 360-pole resolver was chosen for the rate measurement. Both had an angular accuracy of 1 second of one degree. The resolver generated a sine and cosine function for each 2 degrees of rotation. This sine and cosine function from the 12-in. resolver was then further divided into 1/4-second angles by a synchro on a console associated with the test stand. The 7-in. resolver output triggered a time counter and digital printer (Fig. 5). This provided a reading of time in microseconds for each 2-degree rotation of the input axis.

The machining tolerances necessary to produce a fixture compatible with the test requirements required special attention. Some of the areas considered were gimbal orthogonality, input axis shaft friction and wobble, resolver mounting, and stanchion alignment.

The orthogonality requirements of the gimbal bores were not severe. The tilt angle error is a function of the cosine of the angle of non-orthogonality. Thus, a boring error of 5 minutes will result in 0.1 seconds of tilt error. Using optical setups, the boring error can be held to less than 1 minute.

The cone of wobble in the input axis shaft would be seen by the gyroscope as a sinusoidal change in tilt angle input. In the assembly of the completed test

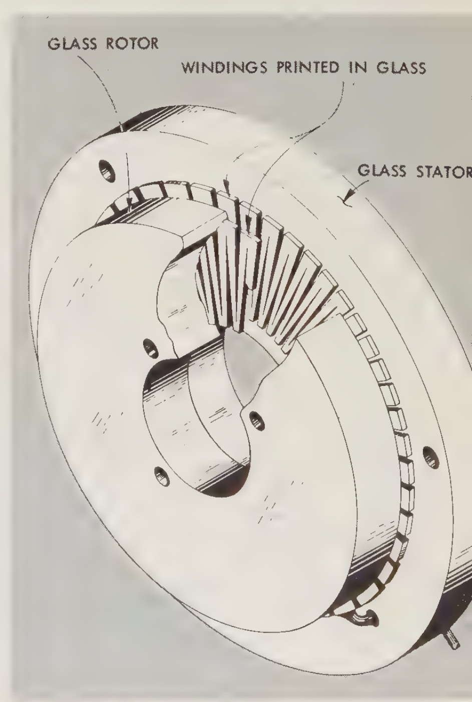


Fig. 4—The test stand uses two resolvers of the type illustrated here. A 12-in. diameter, 360-pole resolver is used for tilt measurement and a 7-in. diameter, 360-pole resolver is mounted at one end of the input axis shaft to measure rate.

stand, this cone of wobble was reduced to an angle of one second by aligning the high eccentricity points of both the inner and outer races of the class 7 duplex bearings.

Concentricity and parallelism of the output shaft bearing assembly were held to 0.001 in. To do this, the input axis shaft was made in three pieces. The gimbal also was a three-piece assembly with the main casting bored through and two precision stub shafts bolted to it. These accuracies are required not only for controlling tilt angle position, but also for reducing the bearing friction and friction variation to a minimum.

Components Selected for High Servo Response

Friction in gyroscopes is significant because changes in friction will affect output and reflect in the table rate as disturbances. Unfortunately, the recorded rate data do not discriminate between gyroscope transients and test stand disturbances.

The slip ring and brush assembly used on the test stand had a uniform frictional drag because of the averaging effect of 72 brush contacts backed up with "soft" springs. Since the brush contact friction was about five times greater than the bearing friction, the total friction was about 3 in.-oz while the brush friction was about 15 in.-oz. These slip rings and brushes, used to transmit power to the rotating shaft and to pick off electrical signals from it, were standard parts made by AC Spark Plug for other systems. These parts were chosen because of their very low electrical noise level.

The d-c torque motor, which drives the input axis shaft, developed a torque of 75 in.-oz at 25 per cent overload. Therefore, with a running torque of about 10 in.-oz, the motor had sufficient torque available for the high response necessary to maintain a tight servo loop.

The choice of the d-c, torque motor gearless direct drive, instead of an a-c servo motor with a gear train, had several advantages. Under production testing, the gear train developed backlash and was a cause of instability in a tight servo loop. The d-c torque motor with its back emf acted as an inertial damper and helped stabilize the loop.

The servo amplifier driving the d-c torque motor was a transistorized design and was mounted in the base of the stand to reduce the length of gyroscope output wiring.

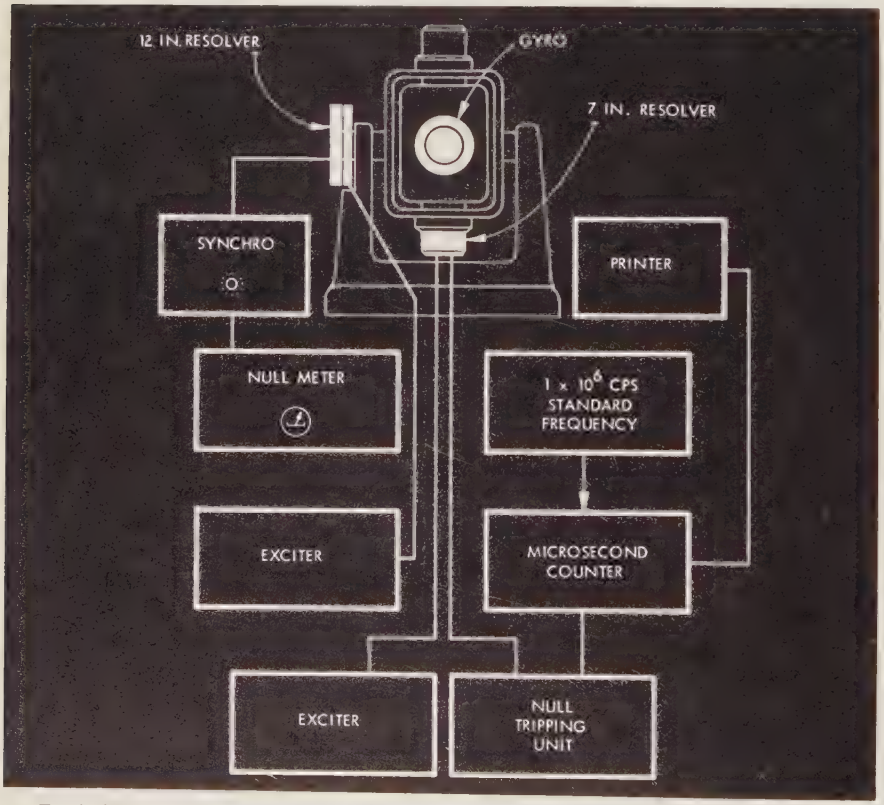


Fig. 5—This block diagram shows the relation of the components of the tilt angle measurement and rate measurement system of the test stand. The rate information from the 7-in. resolver is given by the printer.

Other types of test stands use the principle of a high inertia rotating disc on which to mount a gyroscope. This high inertia is reduced in the AC Spark Plug test stand since it does not have the disc but allows the gyroscope to be placed directly in the input axis shaft for test. The resulting 2,000 in.^{2}-oz moment of inertia of the shaft assembly is four to five times less than the rotating disc design. This lower moment of inertia permits improved servo loop performance.

Conclusion

Currently the new test stand is being used and evaluated. Already it has shown some definite advances in techniques of calibrating gyroscopes.

However, gyroscopes are being improved and new sensing devices are being developed. Since a testing machine must be better than the item being tested, it is obvious that a challenge always exists for engineers to keep pace in their development of testing equipment of any type.

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Planning an Improved Electrical Distribution System for an Automobile Assembly Plant

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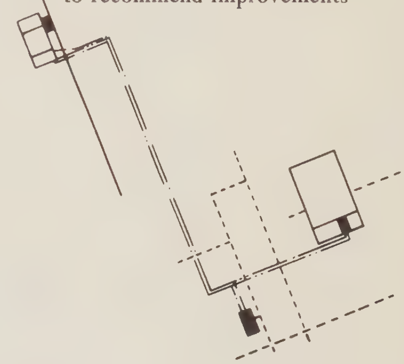
Assembly Division

A prominent area for the application of electrical engineering fundamentals is in power distribution systems for industrial plants. A well designed and well operated system can provide many advantages, such as reasonable operating costs and adequate protection against stoppage of production or damage to processes caused by a power failure. A typical assignment of the electrical engineer is to survey the overall distribution system and load in a plant for the purpose of making recommendations for improved or new systems. An example was a study made recently at the Linden (New Jersey) plant of the Buick-Oldsmobile-Pontiac Assembly Division. This study was necessary because the accumulation of previous load increases, coupled with some new plans for still more increases, indicated that the existing distribution system no longer would be efficient for production requirements. An improved system was proposed using seven sub-stations in a *load-center* arrangement, providing greater capacity, greater flexibility, and allowance for future load increases.

The assignment: study the existing system and loads to recommend improvements



THE automobile assembly plant of the Buick-Oldsmobile-Pontiac Assembly Division at Linden, New Jersey, has increased in floor area by about 30 per cent and has increased in production capacity by about 400 per cent since it was originally opened in 1937. These increases, together with the normal improvements in equipment and methods, have gradually imposed higher demands for electrical power on the plant distribution system.

The capacity of the original power distribution system was 5,500 kva. An addition to the system—a 2,000-kva transformer bank—was made in 1941 raising the capacity to 7,500 kva. However, certain banks in the system were operating at capacity which limited the ability of the overall system to handle emergency overloads and limited further increases in electrical capacity. Therefore, it was decided to survey the existing plant system and the loads and prepare recommendations for improvement and modernization of the system. Reasons for revising the system, in addition to the overload conditions, were: (a) new plans for further expansion of floor space, (b) the need to replace obsolete electrical equipment, and (c) the need to provide for loads that might be added in any future plans.

Surveying the Existing System and Future Loads

Electrical engineers surveyed the system and prepared one-line diagrams of the electrical circuits, layouts of sub-stations, and a layout of the entire plant system (Fig. 1).

The existing electrical loads in the entire plant at the beginning of the study were about 5,000-kva average power and light load and 1,650-kva average welding load. However, the welding load varied from 50 per cent to 150 per cent of this value due to resistance spotwelding operations. Of the total capacity of 7,500 kva, about 5,000 kva was available for power and light and 2,500 kva was available for welding. Power was provided by a *radial* system from four transformer banks arranged in two locations in the plant. (In the radial system, power was distributed from one or two locations to the entire plant by long secondary low voltage feeders).

At one of these locations in the Linden Plant, three transformer banks designated *A*, *B*, and *C* supplied power with a capacity of 2,000 kva each (Fig. 1). Bank *A* operated at about 100 per cent of capacity, Bank *B* at 67 per cent of capacity, and Bank *C* at 118 per cent of capacity. At the second location, one transformer designated *D*, supplied power with a capacity of 1,500 kva and operated at about 67 per cent of capacity. Transformer Bank *B* and approximately 33 per cent of transformer Bank *D* were used for welding loads.

Three-phase, 4,160-volt power was supplied from the utility company's sub-station to transformer Banks *A*, *B*, *C*, and *D* by two sets of parallel aerial feeders. Secondary feeders distributed power to lighting transformers, power panels, and bus duct throughout the plant. In the main building, dry type, single phase, 25, 35, and 50-kva lighting transformers were used to step down the voltage from 480 volts to 220/110 volts. In some outlying buildings, the lighting transformers were dry type, three phase, 208Y/100-volt, 75-kva rating.

While the survey was in progress, a 3,000-kva sub-station was installed to supplement the original radial system and satisfy the more immediate needs for increased power due to the overloaded condition and additional equipment installations. This sub-station, consisting of two 1,500-kva transformer banks supplied with primary voltage, was located near certain loads in another part of the plant. These banks were designated *E* and *F* in the existing system (Fig. 1). Ultimately, these banks were to become a part of the revised distribution system resulting from the recommendations made after concluding the survey.

With the above information available, estimates of future electrical loads and plans for the revised distribution system were made. Some new plans for expansion of the Linden Plant floor space determined the expected load increases caused by power and lighting require-

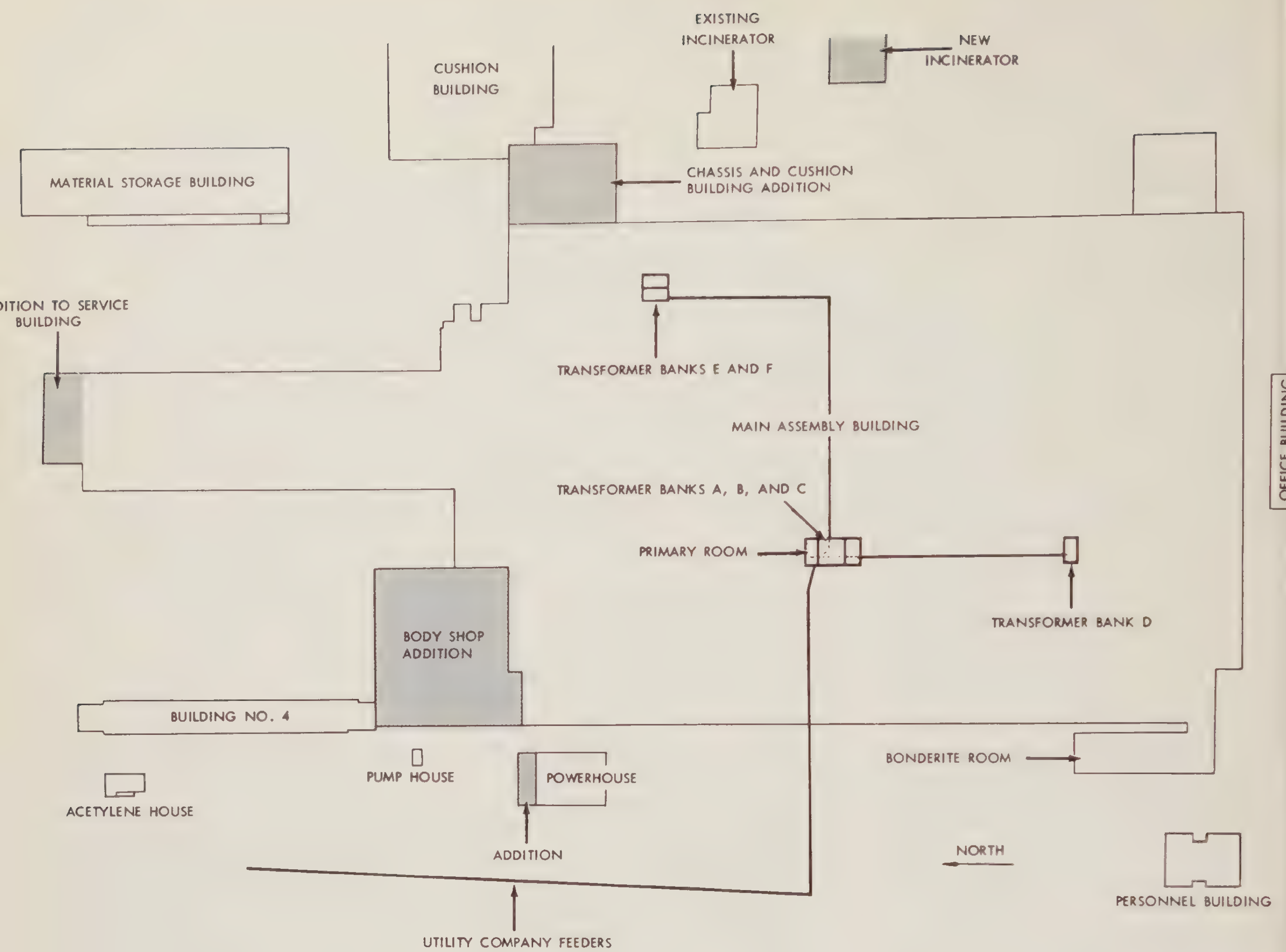


Fig. 1—The layout of the electrical distribution system which existed prior to the study is shown on this plan view of the Linden Plant. When the plant was first opened in 1937, it was served by transformer Banks *B*, *C*, and *D*. Bank *A* was added in 1941. Banks *E* and *F* were added in 1955 and later were to become a part of the revised system. The building additions which figured in the need for a study of the distribution system to improve it are indicated by the darkened areas and are identified as the power house addition, body shop addition, chassis department and cushion building addition, and new incinerator building. Subsequent to this study, a service building addition was made.

ments. The anticipated loads were calculated using a numerical value representing the power and lighting load per square foot of plant area in the existing plant after transformer Banks *E* and *F* were installed. This value was calculated as follows: 5,520 kva ÷ 1,437,615 sq ft = 0.00386 kva per sq ft. (This value of the power and lighting load increased to 5,520 kva from the original 5,000 kva after the installation in 1955 of additional equipment.) The following is a summary of the planned building additions and the corresponding power and lighting loads:

Body Shop Addition—a 31-bay addition to the main building which adds 49,600 sq ft. The electrical load, therefore, was estimated as: (49,600) (0.00386) = 191 kva. No additional welding load was to be installed in this area.

Chassis Department and Cushion Building Addition—a 12-bay addition to the main building which adds 19,200 sq ft. The electrical load was estimated as: (19,200) (0.00386) = 74 kva. No additional welding load was to be installed here.

Power House Addition—a 2,300-sq ft addition to the power house to accommodate an 800-hp, 4,200-cfm air compressor and a new 80,000-lb per hr steam boiler. Equipment loads in this area were estimated as follows: 8.6 kva, 2 sump pump motors (acetylene house waste); 21 kva, 2 cooling tower fan motors; 62.9 kva, 1 forced draft fan motor; 18.1 kva, 1 motor-generator set; 21.4 kva, 3 unit heater motors; 133.4 kva, 4 cooling water pump motors; 3.2 kva, 2 sump pump motors. Total equipment load is 269 kva. Additional lighting load was estimated to be 4 kva. Adding other miscellaneous loads of 20 kva made a total power house load of 293 kva.

New Incinerator—a 6,000-sq ft building to be constructed next to the existing incinerator. The existing incinerator building was planned to remain in use for other purposes; thus, practically

no electrical load would be removed. Motor loads in the new incinerator were estimated to be 227 kva. Power and lighting loads were 23 kva for a total of 250 kva.

The plant expansion, therefore, would add a total load from the above four areas of 808 kva. The original average power and light load was 5,000 kva which was 100 per cent of capacity. The added load of 808 kva due to building expansion plus the 520 kva additional equipment load in 1955 would have raised this percentage to 126 per cent had transformer Banks *E* and *F* not been installed.

Planning the Improved Distribution System

Using information from the survey of the existing distribution system and the proposed new electrical loads, recommendations then were made for a modernized system which would:

(a) Replace certain obsolete equipment and offer safer and more efficient service

(b) Reduce the loading on transformer banks to provide more flexibility for handling emergency overloads in case of temporary failures

(c) Provide an electrical distribution system to allow for future increases in load resulting from processing or equipment changes.

The new system was designed using the *load-center* principle with 2,000-amp, low reactance bus duct. The load-center principle, which utilizes several sub-stations supplied with primary voltage located throughout the plant near loads, was chosen because it has several advantages over the radial system. For example, the load-center system allows plant expansions to be made at lower costs because bus duct can be extended into the new area and loads placed on the circuit breakers of the new bus duct. If substantially higher loads are to be encountered, a new sub-station can be added at the new location. Installation has a minimum interference with normal production operations because primary feeders can be installed on the roof. Expansion of a radial system, on the other hand, usually requires extension of low voltage, high current distribution lines farther away from the single sub-station. This requires, longer, larger size conductors for the same load rating due to the higher currents and greater distances.

Another advantage of the load-center system is a lower voltage drop than in a radial system. This is another result of the shorter secondary voltage distribution lines which produce lower impedance and, therefore, a lower voltage drop. Similarly, power losses are lower in the load-center system because of shorter lengths of distribution lines, lower currents, and lower resistance.

In the secondary power distribution system 2,000-amp, low reactance bus duct was used because:

- It provided flexibility and interchangeability in the overall system
- There was a low power loss and low drop in power factor
- The output of a 1,500-kva bank was 1,804 amperes at 480 volts
- It had lower installation cost than cable in conduit of equivalent capacity to distribute the required power.

On the low reactance bus duct, circuit breakers are installed to feed individual loads throughout the plant. These circuit breakers can be located at any point in the system and are used to isolate various loads from the distribution system as well as to protect the individual load feeders against overloads.

The primary system (4,160 volts) is a selective type system whereby each transformer bank can be isolated from the distribution system, but the load can be shifted onto other transformer banks by two methods:

(a) The load can be shifted onto the adjacent transformer bank where a bus tie section is provided. This is done by removing the secondary main bus breaker from the original transformer section and placing it in the bus tie section which connects the low reactance bus duct to the adjacent transformer bank isolating the bus duct from the other transformer bank

(b) Bus tie sections can be installed between the bus duct in the plant, and therefore, loads can be shifted by opening and closing these sections.

The advantage of flexibility and interchangeability are especially important in automobile assembly plants where the annual model change requires certain revisions to the manufacturing operations and may require changes in the electrical power loads. For example, any load up to transformer capacity can be served equally well from any point on the low reactance bus duct. Also, when a system becomes loaded, another sub-station can be installed between two existing sub-stations and the bus duct sectionalized. This never obsoletes the system; in fact, load increases improve it because the low reactance bus duct becomes shorter as sub-stations are added and, therefore, reduces power losses.

Arrangement of Sub-stations

The new power distribution system proposed for the Linden Plant used low reactance bus duct and seven sub-stations located according to the load requirements in the various plant buildings (Fig. 2).

The first sub-station was located in the power house addition and had a capacity of 500 kva, with provisions for adding 500 kva in the future. It would serve the power and lighting loads in the power house, pump house, building No. 4, and the acetylene house. A double set of feeders would supply the sub-station, so that if one feeder failed, the load could be shifted to the other. The load on the station was 388 kva, or 78 per cent of capacity.

The second sub-station (3,000 kva) was located in an enclosure to be constructed where transformer Bank *D* formerly stood. This sub-station would contain two 1,500-kva transformer banks mounted end to end with a bus tie compartment in the center. It was placed at this site because no rework of building steel was required. Also, this location is in the center of one of the power and light load areas. The station would be supplied by a 4,160-volt feeder from the primary room switchgear.

The second sub-station would supply low reactance bus duct serving the power and light loads in a portion of the main plant plus the office building and personnel building. Bus duct was located along existing column lines which provided the advantages of (a) minimum interference with production, (b) minimum cost of installation, (c) minimum cost of relocating onto the bus, and (d) ease of operation and maintenance of the system.

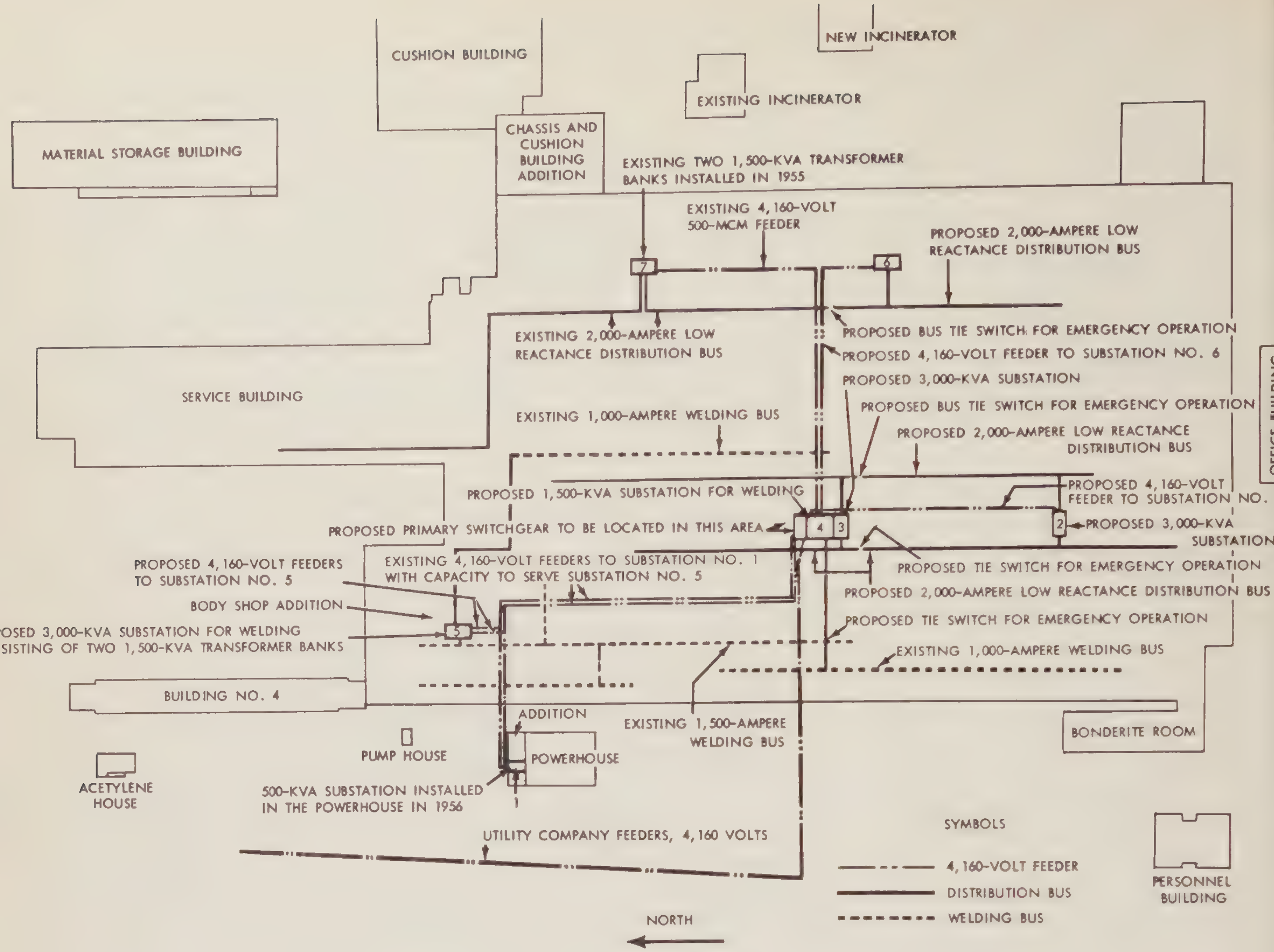


Fig. 2—The study of the distribution system resulted in the proposed load-center system shown in this plan view. It utilizes seven sub-stations supplied by 4,160-volt feeders. Secondary voltage is supplied through low-reactance bus duct. New bus duct is located along building column lines to simplify installation, maintenance, and future relocation. Bus tie switches are used in certain lines to shift the load in case of emergencies. The new primary switchgear room is located at the site of the former transformer Bank *A*. Sub-stations 2, 3, 4, and 7 occupy existing transformer sites while sub-stations 1, 5, and 6 are at new locations. Total capacity of the proposed system is 15,500 kva compared to 7,500-kva capacity obtainable in the original radial system.

The total load on this 3,000-kva sub-station was 1,262 kva. This left room for increase of the electrical load, such as for proposed electric paint ovens in the body painting department.

The third sub-station (3,000 kva) was located in the existing enclosure for secondary switchgear for the previous transformer Banks *A*, *B*, and *C*. The location was selected because no alterations, except removal of the secondary switchgear, would be required before installation was made.

The third sub-station would supply low reactance bus duct serving the power and lighting loads in the north-center and northwest areas of the main building. Bus duct was located along column lines. For emergency operations, bus tie switches would join the two bus ducts supplied by No. 2 and No. 3 sub-stations.

The power and light loads in the area served by the third sub-station were 1,777 kva. The 3,000-kva capacity left sufficient room for load expansion.

The fourth sub-station consisted of one 1,500-kva transformer bank, with provisions for another to be added if required. This bank would be used for welding only, and would serve a 1,000-amp welding bus, with a switch connection to another 1,500-amp welding bus for emergency operation. The sub-station was located on the existing transformer deck, which would require an enclosure to house the new sub-station and primary switchgear. The average welding load was 375 amperes, but it fluctuated from 50 to 100 per cent, or from 155 to 465 kva.

This sub-station provided sufficient capacity for emergency operation and expansion of welding loads.

The fifth sub-station (3,000 kva) also consisted of two 1,500-kva transformers mounted end to end with a bus tie compartment to shift the load in emergencies. A double set of feeders was specified to handle this sub-station and the power house sub-station so that in the event of a feeder failure, both sub-stations could operate on one feeder.

One of the two transformers would serve a 1,000-amp welding bus on which the load varies from 325 to 975 kva. The other transformer would serve a 1,500-amp welding bus where the load varies between 395 and 1,185 kva.

The sixth sub-station consisted of one 1,500-kva transformer bank, with provision for another. It was located to handle power and light loads in the southeast corner of the plant and in the entire east side of the second floor. A tie switch could be installed between the sixth and seventh sub-stations so their loads could be interchanged.

The seventh sub-station in the plans for the revised distribution system was the former 3,000-kva station designated *E* and *F* when used in conjunction with the original system. It was to operate under a load of 2,000 kva and was equally divided between the two 1,500-kva banks. A bus tie switch connected the two banks. One bank provided power and lighting for the northeast corner of the plant, the incinerator, the cushion building, and a material storage building. The other bank supplied the service building.

All of the former primary switchgear was to be replaced when the new system was installed, except for certain circuit breakers which had been installed in 1955. The following new switchgear was to be added: circuit breakers for all incoming primary feeders, and circuit breakers for the second, third, fourth, and sixth sub-stations.

The new switchgear was located in a new enclosure over the transformer deck adjacent to the primary room. This was done to prevent interference with the operation of the former equipment while the new was being installed.

Installation Sequence

The planning of the revised distribution system also included a sequence of installing the new sub-stations and connecting equipment. The procedure was arranged so that power would be supplied continuously to the plant during the alterations and there would be a minimum of interference with normal manufacturing operations.

The first step was to install the first sub-station in a new enclosure on the power house addition. When it was completed, the load would be removed from transformer Bank *C* and shifted to the new station.

The second step was to install the fifth sub-station to serve welding bus duct. After the installation was completed, the entire load from Bank *B* would be transferred to it.

The third step was to shift the entire load from existing transformer Bank *A* onto Bank *B* by reworking the secondary bus. Transformer Bank *A* would be removed from the transformer deck which would be enlarged to become the new primary room enclosure. Primary switchgear then would be added.

The fourth step was to install the sixth sub-station complete with its bus duct system. The loads to be served would be transferred to this sub-station, reducing the loads on transformer Banks *C* and *D*.

The fifth step was to remove the load on Bank *D* by shifting the remaining load to other banks. The welding load would be shifted onto the fifth sub-station by making a tie between the 1,000-amp and 1,500-amp bus ducts. The obsolete transformer bank and secondary switchgear would be removed, and the enclosure enlarged to house two 1,500-kva banks. These banks would be installed along with the entire bus duct system.

After installing the bus duct on the west side of the plant, the sixth step was to close the bus tie switches between the north and south bus sections and shift the loads remaining on Banks *B* and *C* onto the new bus. After the loads were removed from *B* and *C*, the secondary switchgear would be removed and the third sub-station installed in the secondary room and connected to the bus duct system on the first floor. After the sub-station was placed in operation, the tie switches between the north and south bus sections would be opened to remove part of the load from the second sub-station.

The seventh step was to remove Banks *B* and *C* and enlarge the enclosure over the transformer deck. The fourth sub-station would be installed to serve the 1,000-amp welding bus in the south section. After its installation, the welding load would be shifted to it from the fifth sub-station by opening the bus tie switch.

Summary

The electrical power distribution system proposed in this study is an improved load-center system offering the advantages of flexibility, economical operation, more capacity for emergency overloads, and adequate reserve for future expansion.

Another benefit is the replacement of older switchgear and other components with up-to-date electrical equipment.

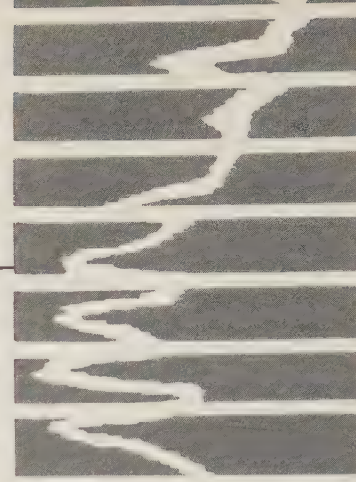
The proposed system provides a total capacity of 15,500 kva compared to 7,500-kva capacity obtainable in the original radial system. Capacity for welding load is increased by 80 per cent from 2,500 kva to 4,500 kva. Power and lighting capacity is raised by about 120 per cent from 5,000 kva to 11,000 kva. If future needs require, an additional 1,500-kva transformer bank can be added in space allowed at sub-station No. 4.

Currently, the Linden Plant has completed the building expansion mentioned above, except for the incinerator, and has converted part of the electrical distribution system according to the recommendations made in the survey. Further conversion work is in progress.

The experience at this plant demonstrates the value in planning for future increases in electrical load. For example, new loads on the system were represented by the following actions which occurred after the survey: installation of more electric paint ovens; expansion of the size of the planned incinerator from 6,000 sq ft to 8,000 sq ft, and enlargement of the service building by 9,600 sq ft.

The conversion of the distribution system has included completion of the 500-kva sub-station No. 1 in the power house with double feeders to accommodate the future sub-station No. 5. Sub-station No. 5 also was provided for in the Body Shop Building addition by constructing a transformer deck. As mentioned, sub-station No. 7 (formerly Banks *E* and *F*) already was installed. The next scheduled work is the installation of all new primary switchgear for the transformer banks except for sub-stations No. 1, 5, and 7. This is necessary because the utility company plans to supply the plant with 26,400-volt power which will require a 26,400/4,160-volt sub-station in the incoming feeder system. The 3,000-kva sub-station No. 5 then will be installed to remove the loads from former Bank *B*. Bank *B* in turn will receive the load from former Bank *A* which will be removed to accommodate new primary switchgear. Thus, as these actions indicate, the first step in the recommended installation sequence has been completed and the second and third steps are underway.

New Studies Provide More Information on Engine Rumble—A Phenomenon of High Compression Ratio Engines



Rumble is a phenomenon associated with high compression ratio gasoline engines. It is a low-pitched rapping noise resulting from abnormal combustion. Rumble is serious from the standpoint that it may limit the extent to which compression ratio can be raised. As a result, much attention has been given in the past few years to the causes and effects of rumble and possible steps that might be taken to minimize its occurrence. Recently, engineers of the Fuels and Lubricants Department of the GM Research Laboratories concluded a study aimed at providing more information on the basic characteristics of rumble, as related to noise and combustion. Also studied were the effects crankcase oils, phosphorus gasoline additives, and certain engine design and operating variables have on rumble. Results of the study showed that the use of phosphorus gasoline additives or certain types of crankcase oil can reduce the occurrence of rumble. Most important, however, was the conclusion that engines having compression ratios as high as 12 to 1 can be operated satisfactorily with respect to rumble if fuels and oils are carefully selected.

In recent years a new type of noise has been heard in high compression ratio engines—engine rumble, a low-pitched rapping noise associated with abnormal combustion. Rumble is distinguished from combustion knock by the lower pitched sound and also by the fact that it usually occurs at higher engine speeds and loads, as encountered in passing or hill climbing. In severe cases, the noise is very loud and might be compared to noise coming from loose connecting rod bearings.

At the present time rumble is recognized as a barrier that must be overcome before substantially higher compression ratios and correspondingly higher engine efficiencies can be reached[1,2].

To gain more knowledge about engine rumble, the Fuels and Lubricants Department of the GM Research Laboratories conducted a study having two objectives: (a) to find out more about the characteristics of rumble as related to noise and combustion, and (b) to determine the effect crankcase oils, fuels, and various engine design and operating variables have on rumble.

Crankshaft Vibration Is Main Source of Rumble Noise

The characteristics of engine rumble noise were investigated by making sound recordings inside the engine compartment of a car having a 12 to 1 compression ratio engine. The recordings were made with the engine running under both normal and rumbling conditions at a speed of 3,000 rpm—full throttle.

Frequency analyses of the recordings (Fig. 1) showed that engine rumble noise resulted from engine vibrations in a frequency range from 500 cps to 1,600 cps. It also was found that rumble had characteristic frequencies of 800 cps and 1,000 cps which did not change when engine speed was varied. This observation indicated that the rumble sound resulted from resonant vibrations occurring in the engine structure. This was confirmed by an investigation of main bearing cap deflection. Strain gages were mounted on the bottom surface of the main bearing caps and oscillograph recordings made of bearing cap deflection during both normal and rumbling operation of the engine (Fig. 2). The recordings confirmed the fact that vibrations exist in the crankshaft main bearing area during rumble which do not exist during normal operation. This was in general agreement with a previous finding[3] which concluded that crankshaft vibration is the main source of rumble noise.

Rumble Caused by Abnormal Combustion

The cause of engine rumble is abnormal combustion, which is related directly to combustion chamber deposits. This has been demonstrated by observing that rumble is eliminated when chamber deposits are removed and that rumble is initiated when deposits are introduced into the carburetor air stream.

To study the combustion process during rumble, an engine was instrumented so pressure development in one cylinder could be observed. A vibration pick up, mounted on the crankcase near the instrumented cylinder, detected the occurrence of rumble. Pressure-time diagrams were recorded during normal combustion and during rumble (Fig. 3).

A study of the combustion process during rumble showed that rumble results from the following sequence of events:

(a) Multiple ignition of the fuel-air mixture by engine deposits
(b) Abnormal pressure development
(c) Shock excitation of engine parts
(d) Resonant vibration producing rumble noise.

Rumble Reduces Engine Efficiency

Noise is not the only objectionable characteristic of engine rumble. Engine efficiency can be reduced as a result of the abnormal and uncontrolled combustion process associated with rumble. Calculations of indicated work based on pressure-time diagrams made during the rumble cycle showed that from 5 to 6 per cent, and as high as 13 per cent, less work was done during the rumble cycle than during the normal cycle.

The lost work is in the form of heat rejected to the combustion chamber walls. The rejected heat may result in overheated and damaged engine parts. However, no engine failures have been encountered that can be attributed directly to rumble.

Rumble Occurrence Measured by Reference Fuels

Tests conducted to study the effect crankcase oils, fuels, and various engine design and operating variables have on rumble were carried out using leaded iso-octane-benzene *LIB* reference fuels to

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Proper selection of fuels, oils may be cure for rumble

measure the occurrence of rumble. (These reference fuels were originally proposed by the Coordinating Research Council for use in deposit ignition studies.)

The LIB reference fuel series is made up of blends of leaded iso-octane in leaded benzene. Each blend contains three milliliters of tetra-ethyl lead *TEL* per gallon. The LIB number of the individual blends corresponds to the percentage of leaded iso-octane in the blend. Since iso-octane has very high resistance to deposit ignition and benzene has very low resistance[4], blends of the two fuels provide a series of reference fuels having varying deposit ignition resistances. For the tests made by the GM Research Laboratories, reference fuels were blended in 10 per cent increments.

The LIB fuels were used to measure deposit ignition, or rumble requirement, of an engine in much the same manner as the octane scale is used to measure antiknock requirement. The *rumble requirement* of an engine is defined as the LIB number of the fuel which produces trace rumble. Knock is not encountered with LIB fuels because all blends have Research and Motor octane ratings of 115 or higher.

10W-30 Oils Have Lower Rumble Ratings

During normal engine operation, crankcase oil is drawn past the piston rings and through the valve guides into the combustion chamber. Although the amount of oil used in this manner is small, the effect it has on deposit ignition and, therefore, rumble occurrence, is extremely important. Certain oils form combustion chamber deposits which cause more rumble than other oils.

To study the relative contribution of commercial crankcase oils to the occurrence of rumble, laboratory evaluation tests were conducted on 10 commercial oils using two dynamometer controlled, 1959 production V-8 engines having a 10 to 1 compression ratio. Equilibrium combustion chamber deposits were accumulated for each oil tested on a light-duty deposit build-up schedule. A commercial gasoline containing 0.22 theories of phosphorus was used throughout the evaluation tests. (One *theory* of phosphorus is the theoretical amount required to convert all lead in the fuel to lead orthophosphate.) Engine rumble requirements during deposit accumulation were determined periodically at an engine speed of 3,000 rpm—full throttle (Fig. 4).

Rumble ratings obtained for the commercial oils tested showed that crankcase oils of 10W-30 S.A.E. viscosity classification had lower rumble ratings than S.A.E. 20W oils (Fig. 5).

Results from laboratory analyses of the 10 oils tested indicated a correlation between rumble ratings and oil volatility characteristics. The best correlation was found when volatility was expressed in terms of per cent residue (Fig. 6). (The residue was obtained from a circulating-air pan evaporation test which consisted of evaporating samples of each test oil at 550°F in a circulating-air oven.) It was found that, in general, oils having a high percentage of residue are more likely to cause engine rumble.

The rumble ratings obtained by the laboratory tests were verified by a road test evaluation of two of the crankcase oils —Brand C, 20 W and Brand F, 10W-30,

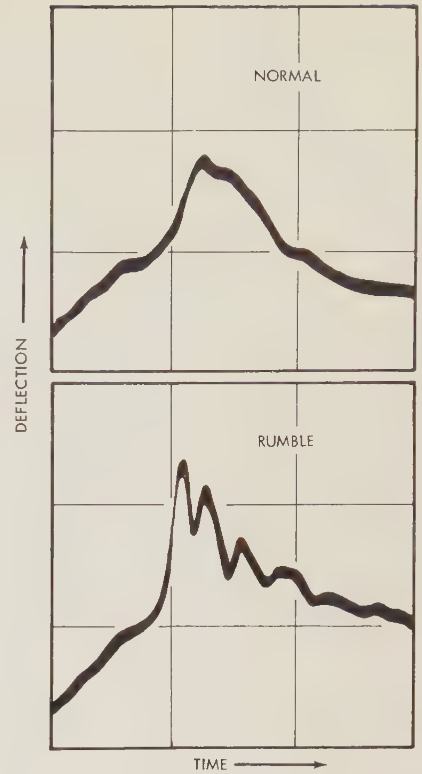

Fig. 2—To confirm the conclusion that rumble noise resulted from resonant vibrations occurring in the engine structure, an investigation was made of main bearing cap deflection during normal and rumbling operation of a 12 to 1 CR engine running at 2,500 rpm—full throttle. Shown here is a typical oscillograph recording of the deflection of the No. 4 main bearing cap during firing of the No. 5 cylinder. As can be seen, vibrations exist in the crankshaft main bearing area during rumble which do not exist when combustion is normal.

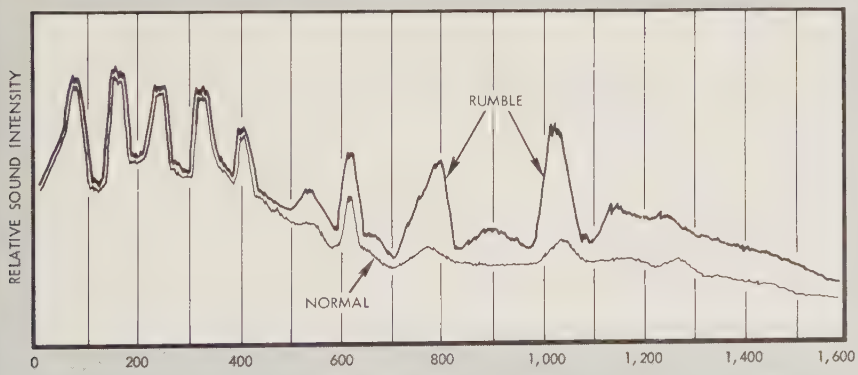

Fig. 1—Characteristics of engine rumble noise were investigated by making sound recordings inside the engine compartment of a car having a 12 to 1 compression ratio CR engine running at 3,000 rpm—full throttle. The engine was operated under both normal and rumbling conditions. Frequency analyses of the recordings were made and are shown here with relative sound intensity as a function of frequency in cycles per second. The upper curve represents the sound analysis for the 12 to 1 CR engine during rumble. The lower curve represents the sound analysis during normal operation. The curves differ principally in two respects: (a) the overall sound intensity for the rumbling engine has a higher level than the normal engine from about 500 to 1,600 cps, and (b) in particular, the sound intensity level is considerably higher at about 800 cps and 1,000 cps for the rumbling engine. These frequencies vary slightly for different engines. As a matter of interest, the frequency of knock for the 12 to 1 CR engine test was about 5,500 cps.

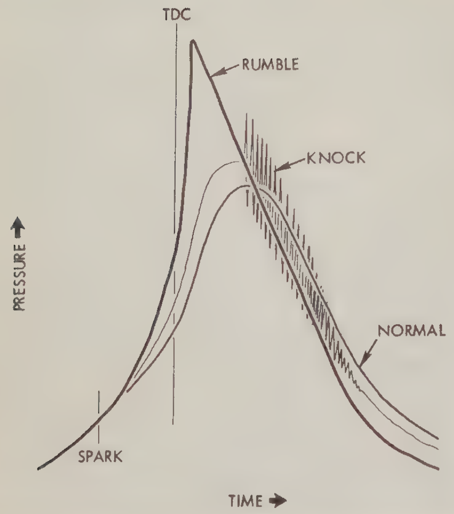

Fig. 3—A comparison of pressure-time diagrams made for a 12 to 1 CR engine running at 3,000 rpm and full throttle under normal, rumble, and knocking combustion conditions showed that cylinder pressure departed from normal very abruptly during the compression stroke when rumble occurred. At this point hot combustion chamber deposits had initiated several flame fronts in addition to the one started by the spark discharge. This gave an abnormally rapid release of energy. (These events would be analogous to igniting the charge with several spark plugs located at various positions in the combustion chamber.) The abnormal characteristics of the rumble pressure-time diagram may be described as follows: (a) a very rapid pressure rise, (b) a very high peak pressure, (c) a sharp peak, and (d) the occurrence of peak pressure earlier in the cycle. Engine cycles having these characteristics cause shock excitation of the engine parts, resulting in rumble. Calculations made of indicated work based on the pressure-time diagram reproduced here showed that from five to six per cent less work was done during the rumble cycle than during the normal cycle.

(Fig. 5). Four cars having the same make and model engine as used in the laboratory tests were used in the road tests. A comparison of the road test results and the laboratory results showed that both tests ranked the oils in the same order. Furthermore, the rumble requirements of the test car engines agreed reasonably well with those obtained for the laboratory engines.

Fuels Affect Rumble

Fuel can affect the occurrence of engine rumble in two ways: (a) through deposits formed in the combustion chamber when the fuel burns, and (b) through the inherent resistance of the fuel to deposit ignition when exposed to high temperatures and pressures in the cylinder during the compression stroke. These two effects must be considered when evaluating the merits of a fuel.

Fuels, as well as crankcase oils, affect the type of deposits formed in the combustion chamber. Deposits from certain types of fuel are more likely than others to cause deposit ignition which can result in rumble. Past investigations have shown that fuels containing very high-boiling hydrocarbons or certain heavy aromatic hydrocarbons form deposits which are more likely to cause deposit ignition and rumble than other fuels (references [5] through [8]). These same investigations and others [9, 10] also have shown that gasoline additives have a marked effect on the deposit ignition tendency of the deposits formed.

To investigate the effect certain types of fuel have on engine rumble requirement, three different fuels were tested in an engine using a commercial 10W-30 oil. The test results (Fig. 7) indicated that: (a) tetra-ethyl lead TEL is a large contributor to engine rumble; (b) the addition of a small amount of phosphorus ignition control compound to a leaded fuel reduces the rumble requirement; and (c) a fuel containing neither TEL nor phosphorus produces no rumble.

From the test results it would appear desirable to remove TEL from gasoline—if rumble was the only concern. From the anti-knock standpoint, however, TEL is very effective and economical. Its use undoubtedly will continue.

The use of phosphorus additives in leaded fuels, on the other hand, appears to be a practical compromise. The anti-knock advantages of TEL can be retained while at the same time deposit ignition and, hence, rumble can be minimized. Phosphorus is a desirable additive because it combines with lead during combustion, forming lead-phosphorus compounds which are less likely to cause deposit ignition than the basic lead chlorides and bromides[11, 12] normally formed when leaded gasoline is burned in an engine.

Phosphorus Additives Reduce Rumble

Nearly all of the phosphorus-containing gasolines sold today contain between 0.1 and 0.3 theories of phosphorus. This amount appears to be reasonably effective for the compression ratios of today's engines. When higher compression ratios are reached, however, this amount may not be adequate.

To investigate the effectiveness of various concentrations of a phosphorus gasoline additive in reducing the rumble tendency of high compression ratio engines, a deposit accumulation road test program was carried out. The phosphorus additive used was a commercial dimethyl-xylyl-phosphate compound. Tests were run using three different phosphorus concentrations—1/2 theory, 3/4 theory, and 1 theory. The base fuel used during the tests with each phosphorus concen-

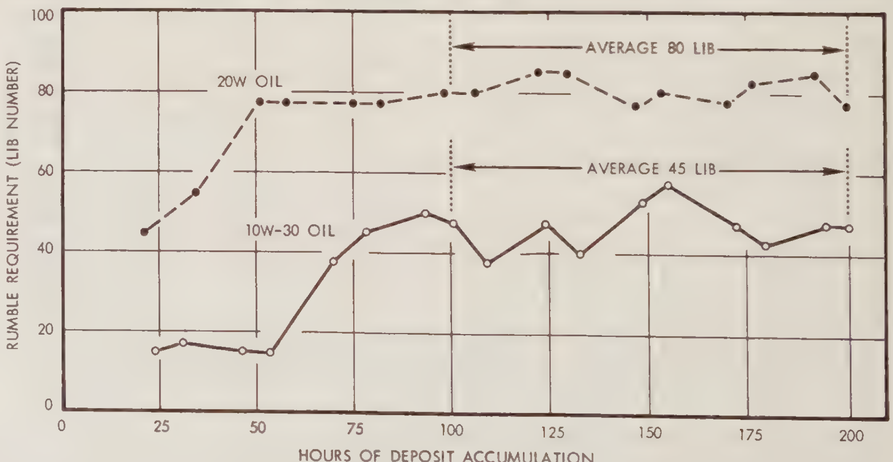

Fig. 4—Engine rumble requirements during deposit accumulation were determined using a dynamometer controlled, 10 to 1 CR engine operating at 3,000 rpm—full throttle. In the chart shown here, engine rumble requirements which occurred with the use of two different crankcase oils are plotted as a function of deposit accumulation time. The individual rumble requirements between 100 and 200 test hours were averaged to provide a stabilized rumble rating for each oil. The higher this rating, the more undesirable the oil with respect to rumble. For example, use of 20W oil rated at 80 LIB is more likely to cause engine rumble than use of the 10W-30 oil rated at 45 LIB numbers. Reproducibility of test results was considered satisfactory since rumble ratings of these two oils repeated within 5 LIB numbers in duplicate tests.

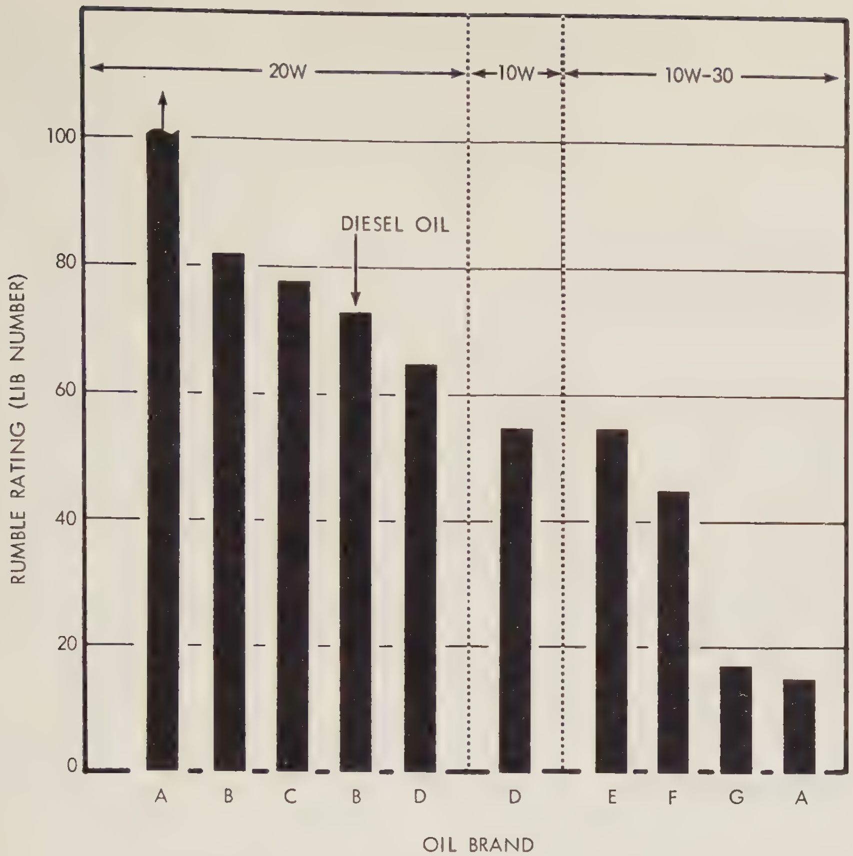

Fig. 5—Rumble ratings obtained for 10 commercial MS crankcase oils of 20W, 10W, and 10W-30 S.A.E. viscosity classifications differed widely. For the five 20W oils, rumble ratings ranged from 65 LIB for Brand D to greater than 100 LIB numbers for Brand A. Ratings for the four 10W-30 oils ranged from 15 LIB for Brand A to 55 LIB for Brand E. The one 10W oil tested was rated at 55 LIB numbers. In every case, the 10W-30 oils had lower rumble ratings than did the 20W oils.

tration was a blend containing 75 per cent alkylate, 15 per cent toluene, and 10 per cent diisobutylene, plus 3 ml TEL per gallon.

The tests were carried out using four cars equipped with special 12 to 1 compression ratio V-8 engines. Cars of two makes were involved. Special pistons and/or cylinder heads, obtained from the engine manufacturers, were used to increase the compression ratio of the engines. Combustion chamber deposits were accumulated under suburban highway driving conditions with vehicle speed limited to 55 mph. LIB rumble requirements, determined at 200-mile intervals, were obtained during full throttle accelerations from approximately 30 mph to 70 mph. All engines used a mid-continent fully distilled 20W oil.

The road tests were arranged so that the three different concentrations of phosphorous were tested simultaneously in different cars during a given time period. After stabilized rumble requirements had been obtained, test fuels containing the various concentrations of phosphorus were switched to different cars. The tests were repeated until each concentration of phosphorus had been tested in each car. Combustion chamber deposits were accumulated for 3,000 to 5,000 miles with each fuel—until a stabilized rumble requirement was obtained. Deposits were not removed from the combustion chambers between tests. Tests also were run to determine the rumble requirement of each car when no phosphorus was used in the fuel.

The results of the road test program (Fig. 8) showed that phosphorus additives in concentrations up to at least one theory can be used effectively to combat deposit ignition and rumble in high compression engines.

Further analysis of the data obtained from the road test program showed that maximum rumble reduction with phosphorus gasoline additives can be obtained only through continuous use. Intermittent use causes the phosphorus to be only partially effective.

Studies Show Need for Fuels With High Deposit Ignition Resistance

The inherent resistance of a fuel to deposit ignition may play an important part in controlling rumble in high compression engines. Fuels having a high resistance will minimize the occurrence of deposit ignition and rumble even in the presence of glowing deposits.

Measurements of the relative deposit ignition resistance of fuels were made as long ago as 1954 in a single cylinder laboratory engine[13]. These studies indicated that a considerable difference exists among the deposit ignition resistance of fuels.

To determine the deposit ignition resistance of present-day commercial gasolines, the Fuels and Lubricants Department developed a technique to rate fuels in a car on the road. Audible rumble was used as an indication of the presence of deposit ignition. Deposit ignition resistance ratings of each test fuel were made by comparing its rumble resistance to that of various LIB reference fuel blends. The LIB rating of a given fuel was determined by: (a) observing the intake manifold vacuum that produced trace rumble during an acceleration when using the test fuel, and (b) determining the LIB blend that also produced trace rumble at the same manifold vacuum. The LIB number of the matching blend represented the deposit ignition rating of the test fuel. The higher the rating, the greater the deposit ignition resistance of the fuel.

A comparison of deposit ignition resistance ratings of fuels both in a laboratory single cylinder engine and in the road tests indicated that the technique was satisfactory.

Having established a workable technique, six commercial super-premium grade gasolines were rated for deposit ignition resistance. The fuels were tested in a 1959 model car equipped with a modified V-8 engine having a 12.5 to 1 compression ratio. The fuel ratings obtained are listed in the following table.

SUPER PREMIUM GASOLINE	AVERAGE DEPOSIT IGNITION RESISTANCE RATINGS (LIB. NO.)
Brand E	57
Brand F	57
Brand G	62
Brand H	57
Brand I	56
Brand J	61

The rating shown in the table for each fuel represents an average of six daily

ratings. The average ratings for the six fuels did not differ significantly in spite of fairly wide differences in hydrocarbon composition.

The ratings for these six fuels indicate that there may be very little difference among the deposit ignition resistances of present-day commercial gasolines. Since none of the six gasolines tested had high resistance to deposit ignition, it appears that a fertile area for further fuel research is the development of gasolines having high deposit ignition resistance. Single cylinder engine studies conducted at the GM Research Laboratories indicate that fuels can be formulated which will have this quality.

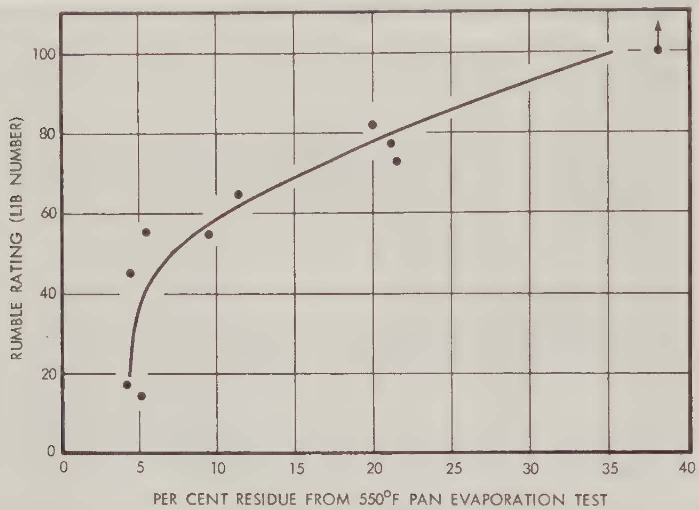

Fig. 6—Results from laboratory analyses of 10 commercial crankcase oils indicated a correlation between rumble ratings and oil volatility characteristics. The best correlation was found when volatility was expressed in terms of per cent residue. The residue was obtained from a circulating-air pan evaporation test which consisted of evaporating samples of each test oil at 550° F in a circulating-air oven. The correlation showed that, in general, the higher the per cent residue, the higher the rumble rating of the oil or, in other words, oils having a high percentage of residue are more likely to cause engine rumble.

Engine Operating Variables Affect Rumble

Engine rumble is a phenomenon that cannot be entirely divorced from engine design. It has been observed, however, that changes in the structural design of an engine, within practical limits, have little effect on the occurrence of rumble[5]. Even though the resonant frequency of the engine parts may be altered, rumble will still occur whenever the rate of cylinder pressure rise or peak cylinder pressures are excessive. Structural design changes, therefore, may alter the characteristics of the rumble noise, but cannot cure the basic cause of rumble—abnormal combustion.

While structural design changes have no effect on the basic cause of rumble, there are certain engine design and operating variables which do because of their influences on the combustion process. These include compression ratio, air-fuel mixture ratio, inlet air humidity and temperature, engine load, and engine speed. Each variable was studied to determine its effect on the occurrence of rumble.

Compression Ratio

The tests conducted to study the effect compression ratio has on rumble show that rumble requirement increases at the rate of approximately 35 LIB numbers per compression ratio (Fig. 9a).

Air-Fuel Ratio

When studying the effect of air-fuel ratio on rumble it was concluded that rumble is most likely to occur at or near maximum power air-fuel ratio. This conclusion was based on test results (Fig. 9b) obtained when using a 12 to 1 compression ratio car equipped with a fuel injection unit. Fuel injection was used to provide uniform mixture distribution and permit changes to be made conveniently in mixture ratio.

Inlet Air Humidity

The pronounced effect inlet air humidity has on the occurrence of rumble was indicated by test results which showed that rumble requirement decreased as absolute humidity increased (Fig. 9c).

Carburetor Inlet Air Temperature

Carburetor inlet air temperature also was found to have an effect on the occurrence of rumble. Tests run on an engine dynamometer under constant absolute humidity and engine coolant temperature conditions showed that a 60°F increase in carburetor air inlet temperature increased rumble requirement by 35 LIB numbers (Fig. 9d).

Engine Coolant Temperature

Engine coolant temperature was found to have very little effect on rumble. A range of coolant temperature from 125°F to 255°F was investigated using a dynamometer controlled 10 to 1 compression ratio engine running at 3,000 rpm—full throttle. Constant values of inlet air temperature and humidity were maintained. The results showed that changes in coolant temperature produced no significant changes in engine rumble requirement. This result was not anticipated, since previous road test experiences had indicated an increase in rumble requirement when coolant temperature increased. It must be remembered, however, that inlet air temperature is partially dependent on engine coolant temperature. When coolant temperature is increased, therefore, any corresponding increase in inlet air temperature may result in a higher rumble requirement.

Engine Load and Speed

Tests conducted to determine the effect of engine speed and load on rumble confirmed the fact that rumble is more pronounced at high engine loads and that rumble occurs more frequently at high engine speeds (Fig. 10).

12 to 1 Engine Operated Under Simulated City Traffic Conditions

Although the effects crankcase oils, fuels, and engine operating variables have on rumble had been determined, there remained one question—would use of "good" fuels and oils allow a high compression engine to be operated without rumble when deposits are accumulated under light-duty city traffic conditions?

To answer this question, two identical cars equipped with 12 to 1 compression ratio V-8 engines were operated under simulated city traffic conditions using a "good" fuel containing one theory of phosphorus. One test car was run with a 10W-30 oil and the second car with a "mediocre" 20W oil. The cars were

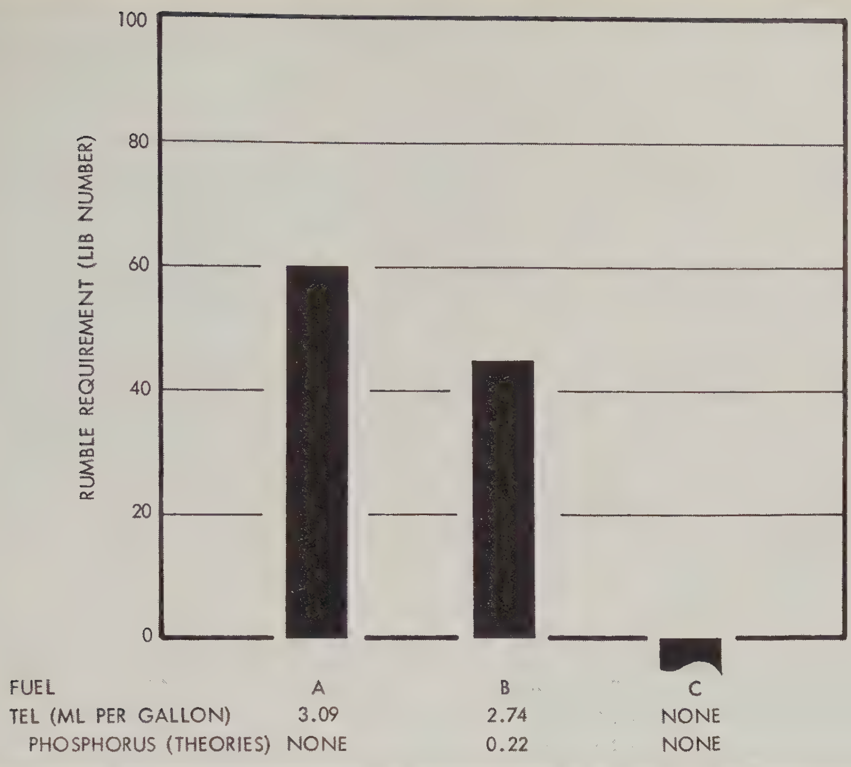

Fig. 7—The effect certain types of fuel have on rumble requirement was investigated using the same test procedure and dynamometer controlled engines as used for the crankcase oil tests. Three different fuels were tested with a commercial 10W-30 oil. Deposits were accumulated with each fuel and average rumble requirements were determined. Fuel A was a gasoline containing 3 ml of tetra-ethyl lead *TEL* per gallon. Fuel B was a commercial gasoline containing approximately 3 ml TEL per gallon plus 0.22 theories of a phosphorus ignition control compound. Fuel C was a commercial gasoline containing neither TEL nor phosphorus. The test results indicated that TEL is a large contributor to the occurrence of rumble, since the engine would not rumble on the lowest LIB fuel when deposits were accumulated with Fuel C. Fuel B, containing a small amount of a phosphorus ignition control compound, reduced the rumble requirement of the engine considerably compared to Fuel A which contained only TEL. These results illustrate the effects of TEL and phosphorus on engine rumble, even though the hydrocarbon composition of Fuel A was somewhat different from that of Fuels B and C.

driven very moderately with top speed limited to 35 mph. The test results (Fig. 11) showed that these 12 to 1 compression ratio engines can be operated satisfactorily with respect to rumble under city traffic conditions if fuels and oils are properly selected.

Summary

Engine rumble is an objectionable noise caused by abnormal combustion. High rates of pressure development in the cylinder, caused by multiple deposit ignition, produce shock-excited resonant vibrations in the engine structure. The result is the low-pitched rapping noise called rumble.

Since the occurrence of rumble is dependent upon the nature of the combustion chamber deposits, any factor affecting deposit composition will, in turn, affect rumble. Tests conducted on crankcase oils to determine their effect on rumble showed a wide difference between engine rumble requirements when using 10 different commercial S.A.E. MS oils. In every case the use of 10W-30 oils gave lower rumble requirements than 20W oils.

The use of high concentrations of phosphorus in the fuel to modify deposit characteristics effectively reduced rum-

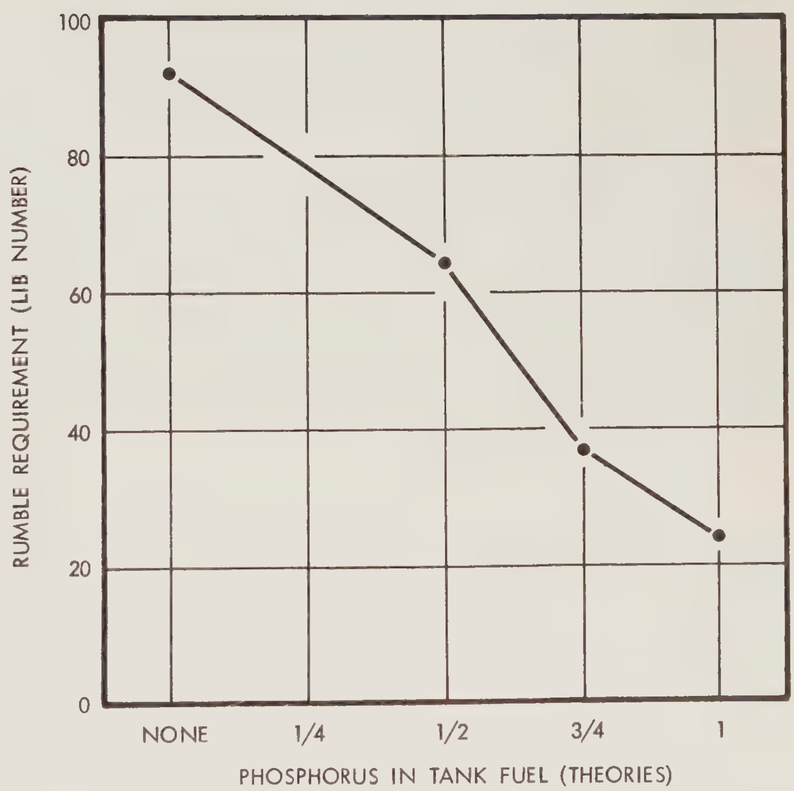

Fig. 8—A road test program was undertaken to study the effect that various concentrations of phosphorus in the tank fuel had on the reduction of rumble requirement. Data for the chart shown here are based on average data taken from four test cars. LIB rumble requirement is shown as a function of the concentration of phosphorus additive used in the tank fuel. Each point represents the average of requirements for all four test cars (10 observations per car). These data indicate that the use of phosphorus in the fuel provides reductions in rumble requirement essentially proportional to the concentration of phosphorus used. It appears that phosphorus additives in concentrations up to at least one theory can be used effectively to combat deposit ignition and rumble in high compression ratio engines.

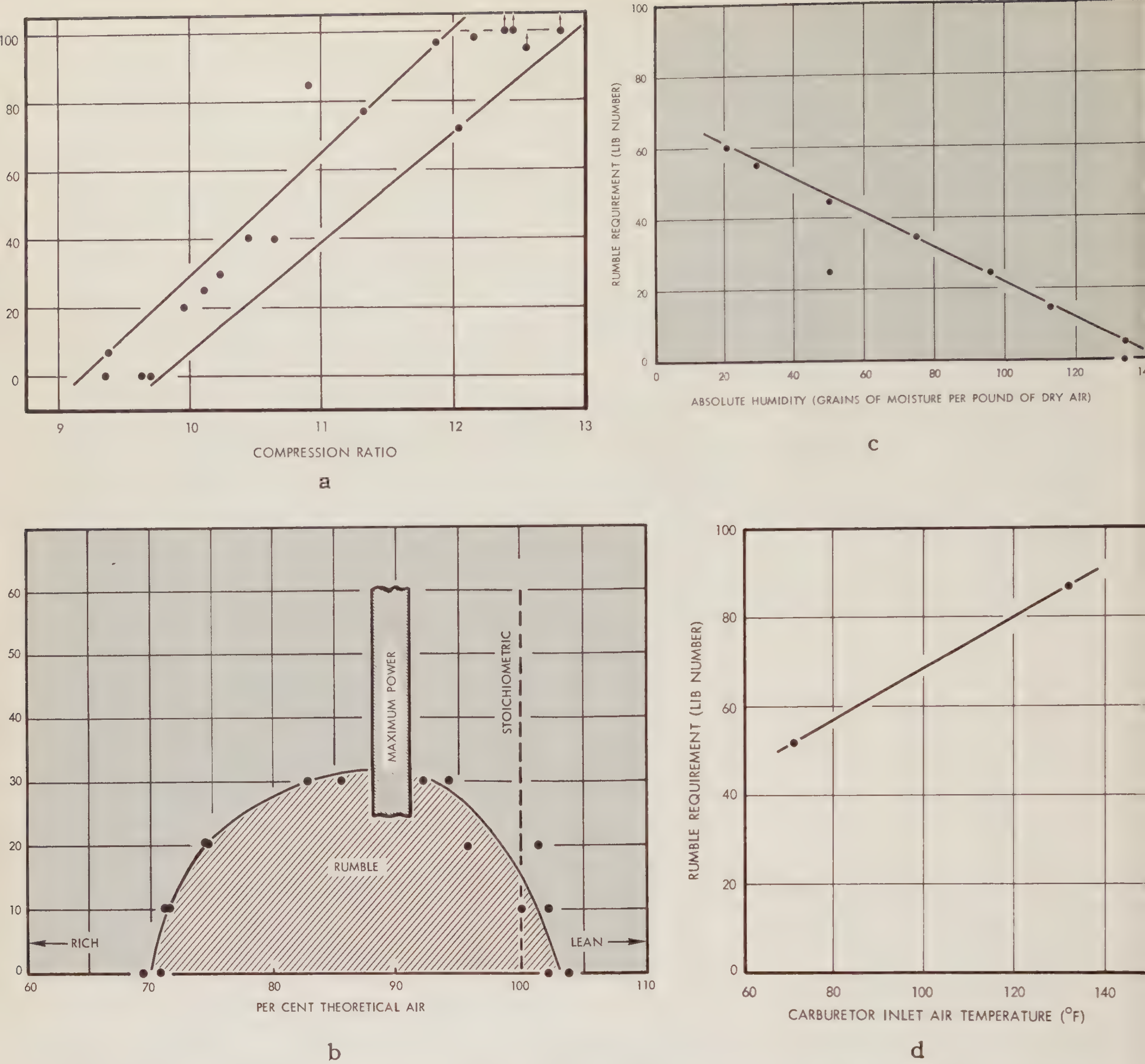

Fig. 9—Certain engine design and operating variables influence the combustion process and, in turn, affect the basic cause of engine rumble—abnormal combustion. To study the effect these variables have on the occurrence of rumble, a series of laboratory and road tests were conducted.

The effect compression ratio has on rumble requirement (a) was studied by road testing cars having production engines and modified high compression ratio engines. The cars were used in company transportation service and accumulated between 1,741 to 30,786 miles. All cars used commercial 20W oils. Cars having production engines were run on commercial premium grade gasolines and cars having the high compression ratio engines used special high octane, non-phosphorus fuels. Rumble requirements were determined during full throttle acceleration from approximately 1,000 to 4,000 engine rpm. With one exception, the rumble requirements fell within a narrow band. The cause for the one unusually high requirement was not apparent. In general, data from the tests showed that rumble requirement increased at the rate of approximately 35 LIB numbers per compression ratio.

The effect air-fuel mixture ratio has on rumble requirement (b) was studied using a 12 to 1 CR engine. The test results were plotted with LIB number requirements as a function of per cent theoretical air in the mixture. Per cent theoretical air was used to express air-fuel mixture ratios because the individual LIB blends have different hydrogen-carbon ratios. The results indicated that rumble requirement is maximum at about 88 per cent to 90 per cent theoretical air, which corresponds to maximum power for the 12 to 1 CR engine used for the test. A mixture of 88 per cent to 90 per cent theoretical air is equivalent to an air-fuel ratio of about 13 to 1 when using commercial gasoline.

The effect of inlet air humidity on rumble requirement was studied in the laboratory using a 10 to 1 CR engine running at 3,000 rpm—full throttle. Results indicated that rumble requirement decreased from 60 to 5 LIB numbers as absolute humidity increased from 21 to 134 grains of moisture per pound of dry air (c). This variation in humidity represented the range normally encountered during a year in the midwest. During the tests, carburetor inlet air temperature and engine coolant temperature were held constant. Humidity was controlled during test observations only.

The influence of carburetor inlet air temperature on rumble requirement was studied under the same test conditions as for inlet air humidity. Test results showed that a 60° F increase in carburetor inlet air temperature increased the rumble requirement by 35 LIB numbers (d). It should be pointed out that the effect of inlet air temperature and humidity tend to cancel each other during seasonal weather variations, since humidity generally increases with ambient temperature.

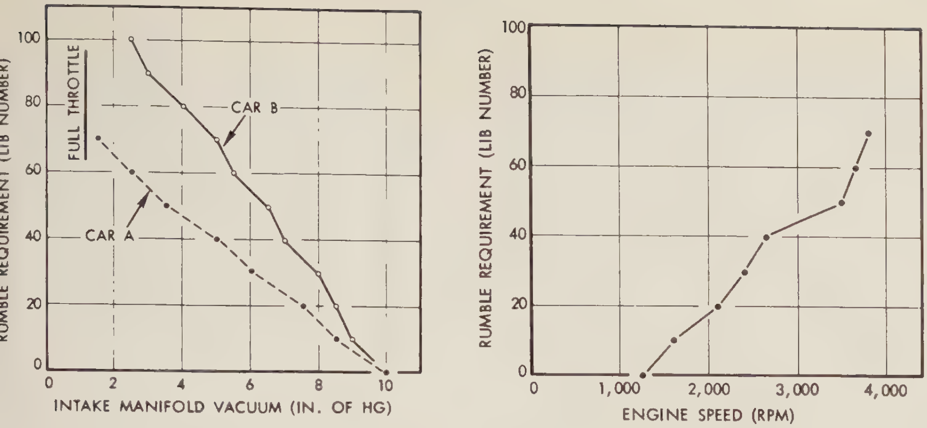

Fig. 10—Both engine load and engine speed affect rumble. To study the effect of engine load, two cars of different makes having 12.5 to 1 CR engines were tested at various engine loads. Rumble requirements were determined during constant manifold vacuum accelerations. The test results showed a reduction in rumble requirement as intake manifold vacuum increased (left). The test results were plotted with engine load expressed in terms of intake manifold vacuum, which is an inverse function of engine load. The effect of engine speed on rumble requirement was determined during full throttle acceleration tests on a car having a 12 to 1 CR engine. The results (right) clearly indicated that engine rumble requirement increases rapidly as engine speed is increased.

ble requirements of high compression ratio engines. Use of phosphorus in concentrations up to one theory gave reductions in rumble requirement essentially proportional to the concentration of phosphorus used. Test results indicated phosphorus must be used continuously for maximum benefit.

A workable technique for rating the deposit ignition resistance of fuels during road tests was developed and used to rate several commercial super-premium grade gasolines. The ratings indicated little difference among the deposit ignition resistances of present-day commercial fuels. The test results pointed out, however, that a desirable area for further research lies in the development of gasolines having high deposit ignition resistance.

A study of the effect certain engine design and operating variables have on rumble occurrence showed that the rumble requirement of an engine is increased if:

- Compression ratio is increased
- Air-fuel ratio approaches best power mixtures
- Inlet air humidity is decreased
- Inlet air temperature is increased
- Engine load and/or speed is increased

The most important conclusion reached by the GM Research Laboratories was that higher compression ratios and correspondingly higher engine efficiencies can be obtained without objectionable rumble if crankcase oils and fuels are carefully selected.

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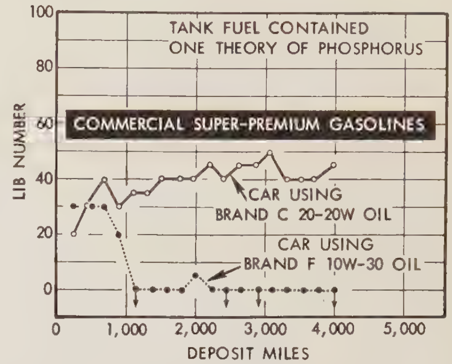

Fig. 11—To determine if a high compression ratio engine could operate without rumble when deposits were accumulated under city traffic conditions, two test cars equipped with 12 to 1 CR engines were driven at moderate speeds. Rumble requirements were determined at 200 mile intervals over a period of 4,000 miles. One test engine used 10W-30 oil and the other used 20W oil. Both engines used a fuel containing one theory of dimethylxylyl phosphate. The rumble requirement for the engine using the 10W-30 oil stabilized at essentially zero LIB. The rumble requirement for the engine using 20W oil stabilized at from 40 to 50 LIB. Superimposed on the chart is a band which includes the average LIB rumble ratings of six commercial super premium gasolines. This band provides an indication of the rumble resistance of commercial-type gasolines for comparison with the test car requirements. The rumble resistance of the six commercial gasolines is higher than the requirements for both test cars. This indicates that 12 to 1 CR engines can be operated satisfactorily with respect to rumble when deposits are accumulated under city traffic conditions if fuels and oils are properly selected.

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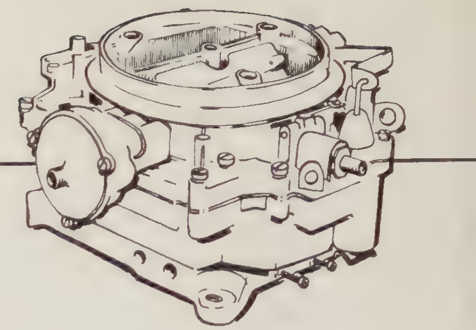
The Flow Box—an Instrument to Measure Fuel Metering Devices for Internal Combustion Engines



The carburetor of an internal combustion engine seemingly has fairly simple jobs to perform: (a) measure and mix the proper quantities of air and fuel, and (b) deliver this mixture to the manifold and engine combustion chamber. In actual practice, however, the carburetor function is complex because of a variety of influences such as driver demands, engine characteristics, or atmospheric conditions. Since these are influences which cannot be predicted by formulas, the engineer's task in carburetor development depends heavily upon laboratory studies, dynamometer tests, and, ultimately, road tests in a vehicle. An important laboratory instrument in this work is the *flow box*, a sealed enclosure which provides a means of measuring the flow of fuel and air through a carburetor being tested. In the flow box, many preliminary studies of a proposed design can be made quickly and accurately without the need for complicated installation on an engine. The flow box is a tool which enables the engineer to do a better job of designing quality carburetors to meet the changing requirements brought on by improvements in motor vehicles.

The pioneers of automotive development recognized at an early date a need for a laboratory instrument capable of measuring carburetion performance. About 1920, such a device was developed. Essentially it consisted of a large metal box in which a carburetor was mounted. The box was sealed so that a measured quantity of air was permitted to flow into it and through the carburetor. At the same time a measured quantity of fuel was introduced to the carburetor. These measurements of fuel and air, when plotted in the form of a graph and shown as air-fuel ratio, enabled the automotive engineer to observe carburetor performance. While techniques have been improved and design refinements made, the basic principle of the first flow box built shortly after World War I still is used in laboratories today.

The automotive engine fuel-air requirements dictate optimum design of a fuel metering device for a given engine. To properly fulfill at least its basic purpose, the fuel metering device must deliver to the intake manifold a mixture of fuel and air in the proper quantity and quality, chemically and physically, so that it can be easily and readily controlled, distributed equally to all cylinders, and consumed by the engine. To accomplish this the fuel metering device must:

- Accurately measure the amount of air drawn in by the engine, and meter the fuel into the air stream in proportion to this quantity of air
- Assist distribution by discharging the liquid fuel into the air stream in the form of an atomized spray
- Provide a means of controlling engine power output and speed in accordance with operator demands by varying the quantity of this mixture of air and fuel, which is permitted to enter the engine.

It has several other equally important functions, but this paper is principally concerned with the three functions listed above.

Some Carburetor Fundamentals Which Apply in Flow Box Studies

Engineers know that gasoline engines will tolerate a spread in air-fuel mixtures varying from lean in the order of 19 to 20 lb of air per lb of fuel to rich in the order of 7 to 8 lb of air per lb of fuel. Obviously, best operation is obtained over a much narrower range.

The chemically correct mixture is the one in which all of the carbon and

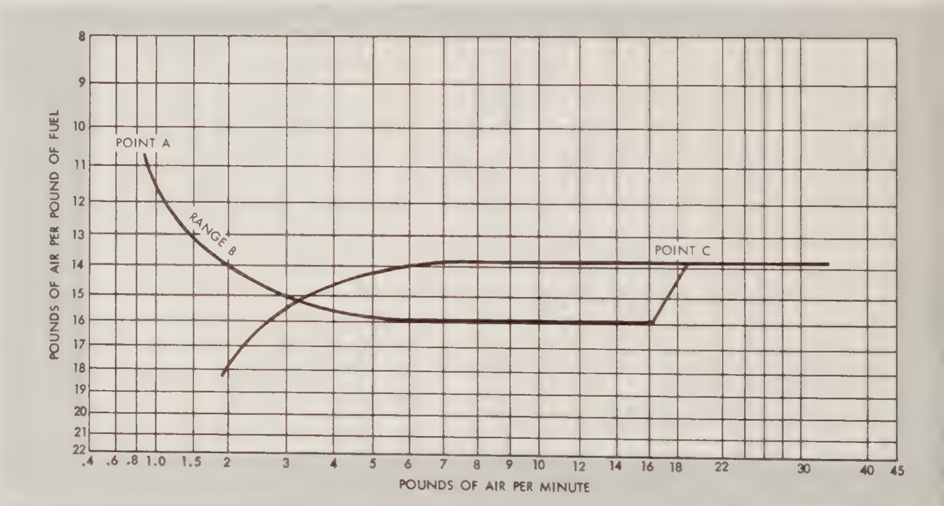


Fig. 1—The performance of a carburetor in a flow box test is indicated in a *flow curve*. This typical flow curve shows that at the idle condition, *Point A*, the engine is using less than one lb of air per min and the air-fuel ratio requirement is about 10.6 to 1. It shows that as the throttle gradually opens, *Range B*, exhaust gas dilution becomes less so that as the curve approaches the part throttle or maximum economy condition, the mixture ratio has been gradually leaned out to about 16.0 to 1. At 19 lb of air per min, *Point C*, the throttle is nearly wide open and the mixture has been enriched two or three ratios for maximum power. The upper curve is called the wide open throttle curve. The drop in this curve to the left indicates a drop in the manifold vacuum caused by a decrease in engine speed when a vehicle is traveling up a constantly increasing grade. Test conditions in the flow box usually are at 72°F room temperature, 45 per cent relative humidity. Fuel temperature is 72°F, fuel inlet pressure is 6 psi, and the fuel specific gravity is 0.735.

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Rochester Products
Division

Engineers use flow curves

to clarify complex

function of carburetor

hydrogen in the fuel combines completely with all of the oxygen in the air when the mixture burns in the cylinder. For perfect combustion, approximately 11.6 lb of air is required to completely burn one lb of carbon. Similarly, 34.6 lb of air is required for one lb of hydrogen.

A typical automotive gasoline contains about 86 per cent carbon and 14 per cent hydrogen by weight. Therefore, the air requirement for theoretically perfect burning of one lb of gasoline is in the order of 14.8 lb, or the chemically correct air-fuel ratio is 14.8 to 1. This ratio, of course, varies slightly with fuel composition. It is apparent, therefore, that there must be, for any given fuel, an air-fuel ratio where all of the fuel reacts with all of the air available and the combustion process liberates a maximum amount of energy. If more air is used so that this ratio becomes leaner than the chemically correct one, excess air will be left after the burning process is completed, but all of the gasoline will have been consumed. If excess gasoline is used so that this ratio becomes richer than the chemically correct one, all of the air will be used but there will remain unburned gasoline at the end of the burning process.

At first glance it would appear, therefore, that the fuel metering device has but a relatively simple job to perform. It merely has to mix one lb of fuel to every 14.8 lb of air that it discharges into the combustion chamber. Unfortunately, perfect conditions do not exist for the following reasons:

- Varying amounts of exhaust gases remaining in the cylinder dilute the fresh charge
- The fuel and air are not perfectly mixed
- Changes in the fuel metering device do not result in equal changes in the mixtures delivered to the combustion chambers by the manifold
- The intake manifold does not deliver exactly equal air-fuel mixtures to all cylinders
- The user of the vehicle is not always interested in maximum engine power.

The power-producing ability of an engine is dependent upon the amount of air it can consume. Hence, when maximum power is desired, it is necessary that all of the oxygen in the mixture be burned. Therefore, the fuel metering device should add enough gasoline so that even with imperfect induction, every molecule of this air will be used in the combustion process. It is evident, therefore, that the mixture ratio for maximum power must be something less than the chemically correct ratio and usually is in the range between 12.5 and 14.5 to 1, as measured by the flow box.

For maximum economy, it is necessary that every particle of fuel find a particle of oxygen, so that complete combustion of the fuel will result. Because of imperfect induction conditions, additional air is required to insure that each molecule of fuel finds its necessary particle of oxygen. As a result, the individual cylinder air-fuel ratio for maximum economy is somewhat leaner than that calculated for chemically perfect combustion. It should be noted, however, that the fuel metering device may be required to deliver richer than chemically correct mixtures to overcome the effects of unequal distribution among the cylinders. Generally speaking, however, economy mixtures at the fuel metering device will range between 14.5 and 17 to 1, as measured by the flow box.

At idle and relatively small throttle openings, very little air and fuel are permitted to enter the engine. Because of valve overlap and the fact that the pressures in the cylinders are extremely low, all of the products of combustion cannot be exhausted from the combustion chamber. Therefore, even though a nearly chemically correct mixture ratio might leave the fuel device, it would be

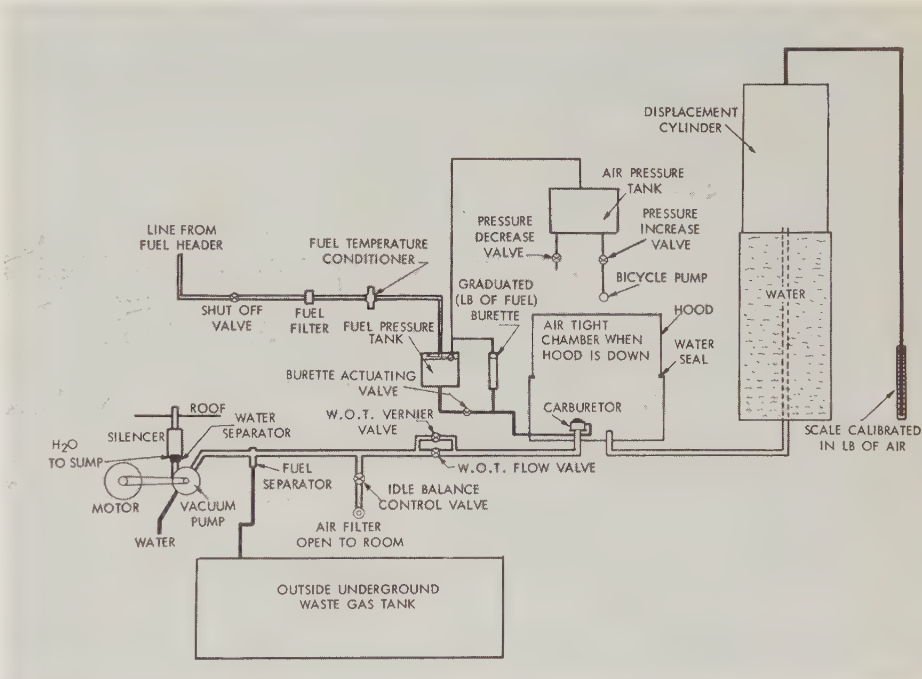


Fig. 2—Shown above is a schematic diagram of the *displacement* type flow box, so named because of the method used to measure air flow through the fuel metering device being tested in the flow box. A measured volume of air within the displacement cylinder constitutes the air supply for a single test in the flow box. As this volume of air flows through a standpipe and through the hood, the displacement cylinder moves downward. This movement is calibrated in terms of pounds of air. By timing the movement, the flow in lb of air per min is obtained. The vacuum pump provides the air flow. Fuel is supplied to a pressurized storage tank from which it is metered at the designated temperature and pressure for the test. Two types of measuring instruments are used to meter the fuel supply: a burette for low fuel rates, and a flow meter (not shown) for high fuel rates.

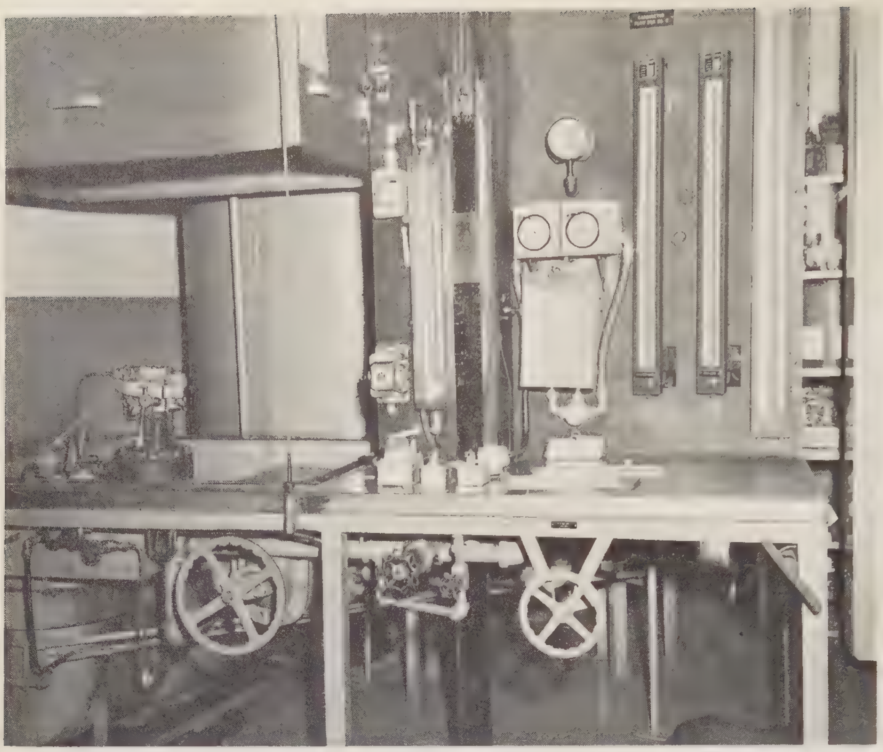


Fig. 3—A typical flow box of the displacement type is shown above with a carburetor installed in a test position. The upper part of the box (hood), shown in a raised position, is lowered and fits in a water trough seal during a test.

diluted in the combustion chamber with exhaust gases until the resultant mixture ratio might be leaner than 20 to 1 and hence not support combustion. Thus, it is necessary for the carburetor to provide an air-fuel mixture that is considerably richer than that required for maximum economy operation.

In carburetor development, the engineer applies principles such as those discussed above to various designs of metering devices. To measure the effectiveness of a proposed design, the metering device is tested in the laboratory flow box and the information about its performance is presented in the form of graphs termed *flow curves* (Fig. 1). Flow curves represent the ultimate objective of flow box testing. They provide the engineer with accurate data for design evaluation.

Two Types of Flow Boxes Aid Carburetor Development

The flow box provides the means by which fuel and air can be measured under laboratory conditions. It is not, however, a substitute for dynamometer testing or vehicular road testing of carburetors because the flow box is incapable of simulating engine pulsations and manifold distribution. But, the flow box does:

- Provide a rapid means of measuring metering characteristics
- Permit analysis of the various metering systems on a carburetor or fuel injection unit
- Reduce the use of engines for experimental tests and possible damage to them by virtue of improper metering
- Eliminate involved installation which would occur on a test engine.

Rochester Products Division uses two types of flow boxes which are classified according to their principle of air measurement: (a) quantitative or *displacement* type, and (b) *orifice* type. The flow boxes are located in rooms which have controlled temperature and humidity. Air flow readings are adjusted to compensate for variations in barometric pressure.

Displacement Type—Air Measurement

For air measurement, the displacement box uses two concentric, vertical cylinders, each closed at one end. The cylinders are located separately from the flow box hood (Fig. 2). One cylinder is stationary and is open at the top; the other cylinder is open at the bottom and moves inside the stationary cylinder thus forming a tank in which the volume can vary. The movable, or upper, cylinder fits into a circular water well around the inside of the stationary cylinder to seal the volume between the two cylinders. An access pipe enables air to flow in either direction between this volume and the flow box hood, causing the upper cylinder to rise or fall. The pipe is

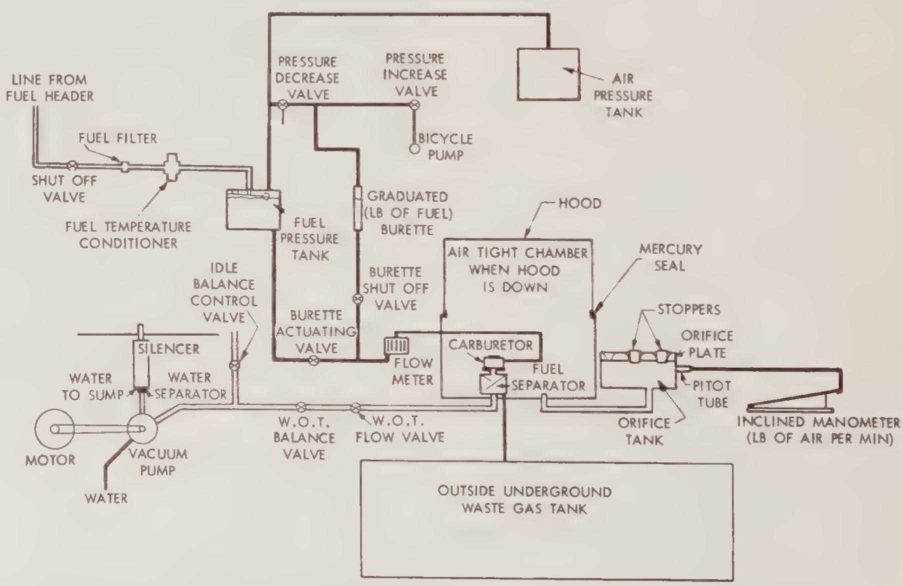


Fig. 4—The *orifice* type flow box differs from the displacement type primarily in the method of air measurement. Air is measured by drawing it through one or more orifices and then into the flow box. A pressure tap mounted in the orifice tank senses the air flow which is measured by an inclined manometer calibrated to read directly in lb of air per min. The orifice type box has several advantages over the displacement type such as permitting faster testing procedure and permitting a continuous volume of air to flow—as opposed to a fixed volume in the displacement type. Air measurement in the orifice type, however, is not as accurate as in the displacement type.

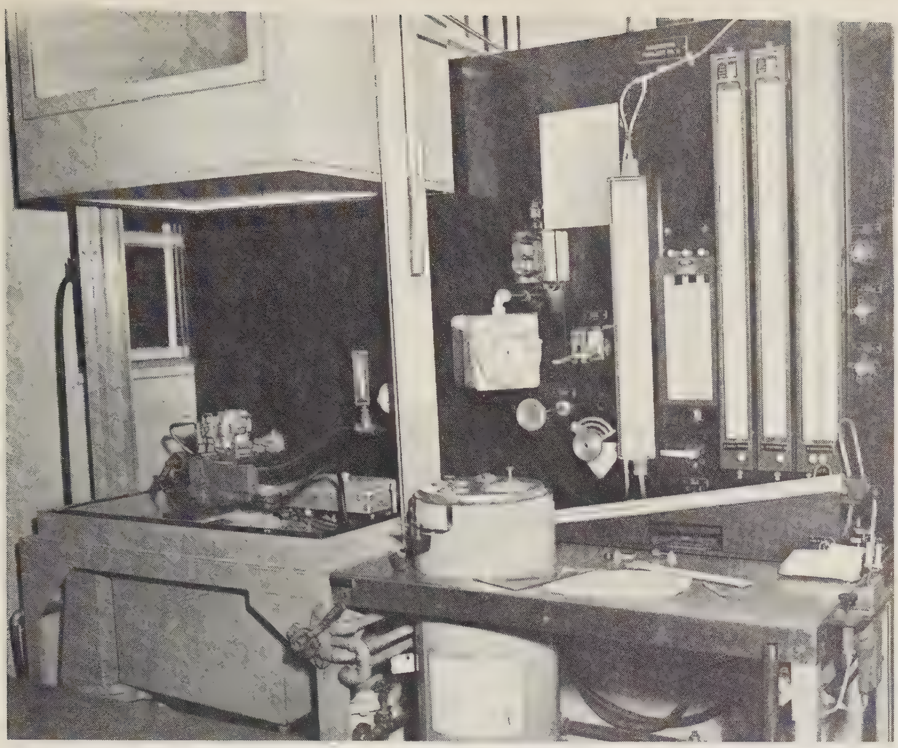


Fig. 5—An orifice type flow box is shown above with a carburetor installed in a test position. The orifice tank and inclined manometer are seen in the foreground.

large enough to minimize line loss and its effect on air flow measurement.

The upper cylinder is counterbalanced and is relatively free of friction. A very small change in pressure causes the upper cylinder to move. Movement of the upper cylinder is calibrated in terms of pounds of air and is shown on a vertical scale. This movement is timed and the resultant calculation is expressed in pounds of air per minute.

When the movable cylinder descends and air flows from it, the liquid height of the water seal increases because of displacement of water by the movable tank. This inherent error in measurement of the displaced volume is overcome by maintaining a constant stream of overflow water to hold the liquid level at a fixed height.

It is obvious that the size of the upper cylinder limits the duration of the flow box test as well as the size of the fuel metering device that can be tested. The basic advantage of this construction, however, is the accuracy of the air measurements at low air flow rates.

The flow box hood is constructed in two sections (Fig. 3). The bottom section of the hood assembly is fixed and houses a plate on which the fuel metering device is mounted. The upper section of the hood assembly may be moved up or down and is counterbalanced for ease of operation. When at the bottom of its travel, the upper hood fits into a water trough around the bottom section sealing the enclosure over the fuel metering device. This upper hood assembly can be raised to:

- Permit changing of the fuel metering device that is to be tested
- Manually adjust the throttle of the fuel metering device
- Permit room air to flow through the hood assembly and into the standpipe which leads into the displacement cylinders, thus recharging these cylinders for a subsequent air flow measurement.

Pressure differential is provided by means of a water sealed constant-speed turbine pump connected by a pipe to the base of the carburetor. The pump is capable of prolonged operation at high vacuum.

The vacuum line between the base of the hood assembly and the pump is equipped with one main adjustable valve and a vernier valve. These two valves are designed to control maximum air flow (wide open throttle). A bleed off valve (idle balance control valve) bleeds atmospheric air to the pump, thereby controlling idle vacuum at the carburetor.

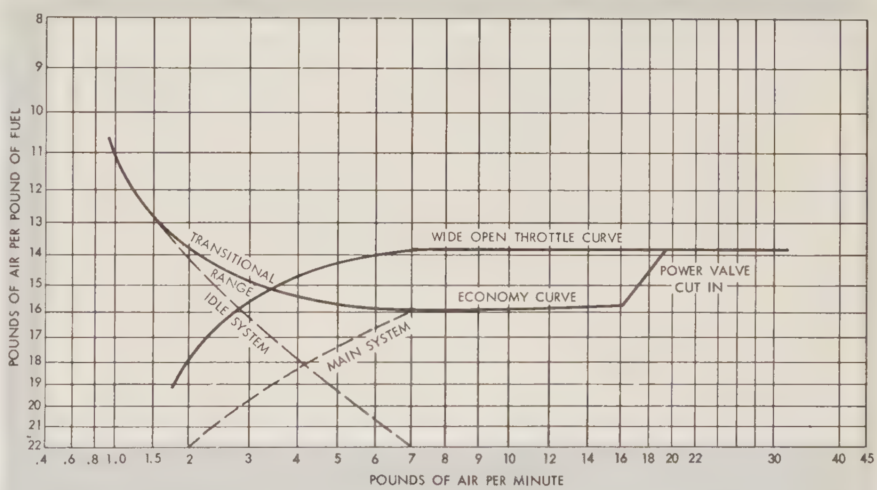


Fig. 6—A typical study in carburetor development may result in flow curves as shown above. The air flow at engine idle, in this example, is 0.9 lb per min. Separate curves for the idle system and for the main metering system of the carburetor are plotted as shown. The transitional range curve represents the simultaneous flowing of both systems. At a flow of 7 lb of air per min, the economy curve begins and continues to 16 lb of air per min where the power valve of the carburetor opens and provides enrichment up to a flow of 19 lb of air per min. As the throttle valve is fully opened, this carburetor reaches its maximum air rate of 34 lb of air per min with a 13.8 to 1 air-fuel ratio, indicated by the wide open throttle curve. The engineer studies performance characteristics such as these in the development of improved carburetors and improved engine designs.

The main valve, vernier valve, and atmospheric bleed valve are adjusted to balance the pressure drop across the fuel metering device at both idle throttle position and wide open throttle. This technique is known as *balancing the box* and must be accomplished before beginning a test. The air flow regulating

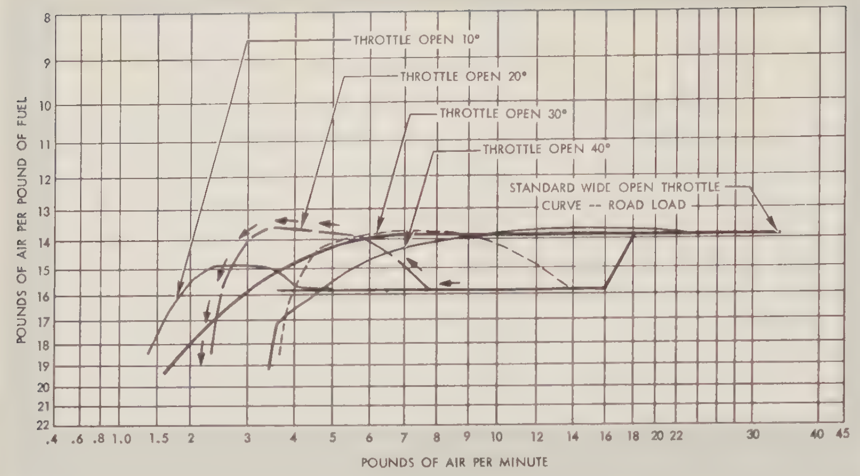


Fig. 7—Typical curves which are studied in carburetor development are *crowd* curves. This graph shows four different crowd curves representing four fixed throttle positions of a carburetor while the manifold vacuum decreases to simulate uphill climbs at fixed throttle. The standard wide open throttle, road load curve also is shown. Each of the crowd curves follows the economy curve until the manifold vacuum reaches the power valve cut-in range, at which point the air-fuel ratio becomes richer and finally falls off to the equivalent of a stall condition. In this example, the fact that each curve is smooth and free of dips or humps means that the engineer can be reasonably certain that this device will have no abnormalities in lean or rich conditions for this uphill climb condition. This information also means that the vehicle operation will show no signs of surge. (*Surge* is the effect of uneven variance of air-fuel ratio and indicates a sudden lean or rich condition resulting in loss of or sudden gain in power. The driver may refer to surge as a "bucking" sensation.)

valves are set in accordance with standard practice to produce a pressure drop of 18 in. Hg when the fuel metering device throttle valve is adjusted for idle air flow. The throttle valve is then opened to its maximum flow position and the pressure drop across the unit is set to either 1 in. Hg or 3 in. Hg, depending on the air flow capacity of the particular engine under study. A recheck and usually some readjustment of the air bleed valve is necessary before the box can be considered balanced.

The values of 18 in. Hg at idle, and 3 in. Hg or 1 in. Hg at wide open throttle, are arbitrary figures selected because they are representative of actual engine operating conditions. However, it is recognized that both idle and wide open throttle positions may result in different vacuum readings as determined by the operating conditions of the engine.

Displacement Type—Fuel Measurement

For fuel measurement in flow box tests, the fuel is pumped from an underground tank to a *header* located above the air box. It then flows through a temperature conditioning device which conditions the fuel to 72°F. The fuel is then fed to a pressurized storage tank to assure an adequate supply of fuel at a fixed temperature and at the desired pressure.

To assure accurate fuel measurements, two types of measuring devices are used. At low fuel rates, a burette is used with a manual timing mechanism. For high fuel rates a flow meter is used.

Fuel is metered and atomized into the air stream by the device being tested. The fuel is later separated from the air stream by a baffle type separator and collected in an underground waste fuel tank. The air is discharged to atmosphere.

Orifice Type—Air Measurement

The orifice type flow box used at Rochester Products Division differs from the displacement type primarily with respect to the air measuring and control systems (Fig. 4). The fuel measuring and vacuum systems of both types of boxes are similar.

In the orifice type, conditioned room air is introduced into the box through orifices which have been machined into a plate and mounted in the top of an orifice tank. One or more of the orifices may be closed by stoppers, thus permitting the flow of air to be varied over a range from 0.2 to 45.0 lb per min.

Consideration has been given to both the thin plate orifices and the flow nozzle, also known as the smooth edged or round edged orifice. While the latter is more difficult to manufacture than the thin plate orifice, it is used because it is less vulnerable to damage and therefore decreases the possibility of inaccuracy in the air flow measurements.

Air flows through the open orifices

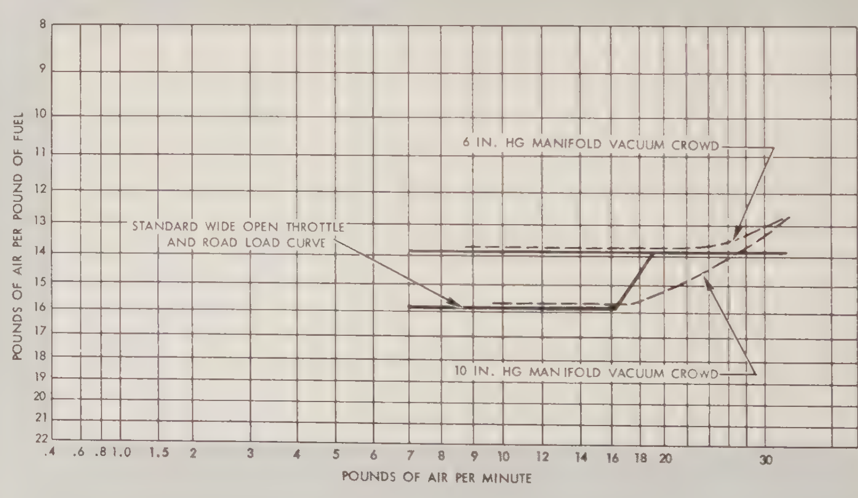


Fig. 8—Additional crowd curves are shown on this graph representing the condition of constantly increasing acceleration on a level road in which the throttle is opened steadily and the manifold vacuum is maintained at a given value. In the flow box tests for this type of operation, the desired flow characteristics may be studied with the power valve opened (at 6 in. Hg) and with the power valve closed (at 10 in. Hg). These tests enable the engineer to investigate the carburetor performance just before and after opening of the power valve. In the carburetor design, it is important that the curve indicate freedom of a lean condition on the 10-in. crowd (just before power valve opening) and freedom of too rich a condition on the 6-in. crowd (just after the power valve is fully opened). Too lean a condition on the 10-in. crowd would cause a surging condition and could result in engine damage from overheating. Too rich a condition on the 6-in. crowd would result in loss of economy.

into the orifice tank and through a pipe into the hood assembly (Fig. 5). As in the case of the displacement type box, air is drawn through the carburetor by the vacuum pump. Air flow is sensed by a pressure tap in the orifice tank and its signal is transmitted to an inclined manometer calibrated to be read directly in pounds of air per minute.

The principal advantages of an orifice type over a displacement type box are:

The Real Results: Fuel and Air Flow Curves

The ultimate objective of flow box tests is to obtain data which relate to fuel and air flow. Flow curves present these data in graphical form in terms of the relationship of air-fuel ratio to air flow. Air-fuel ratio is the ordinate of the curve (expressed in lb of air per lb of fuel), and air flow is the abscissa (expressed in lb of air per min).

starts delivering fuel at 2 lb of air per min and continues enriching until it reaches the road load, or *economy curve*, at 7 lb of air per min.

The transition range represents the simultaneous flowing of the idle and main metering systems in which the two systems must blend smoothly.

Proceeding with the analysis, the *economy curve* continues to 16 lb of air per min at which point the power valve* opens.

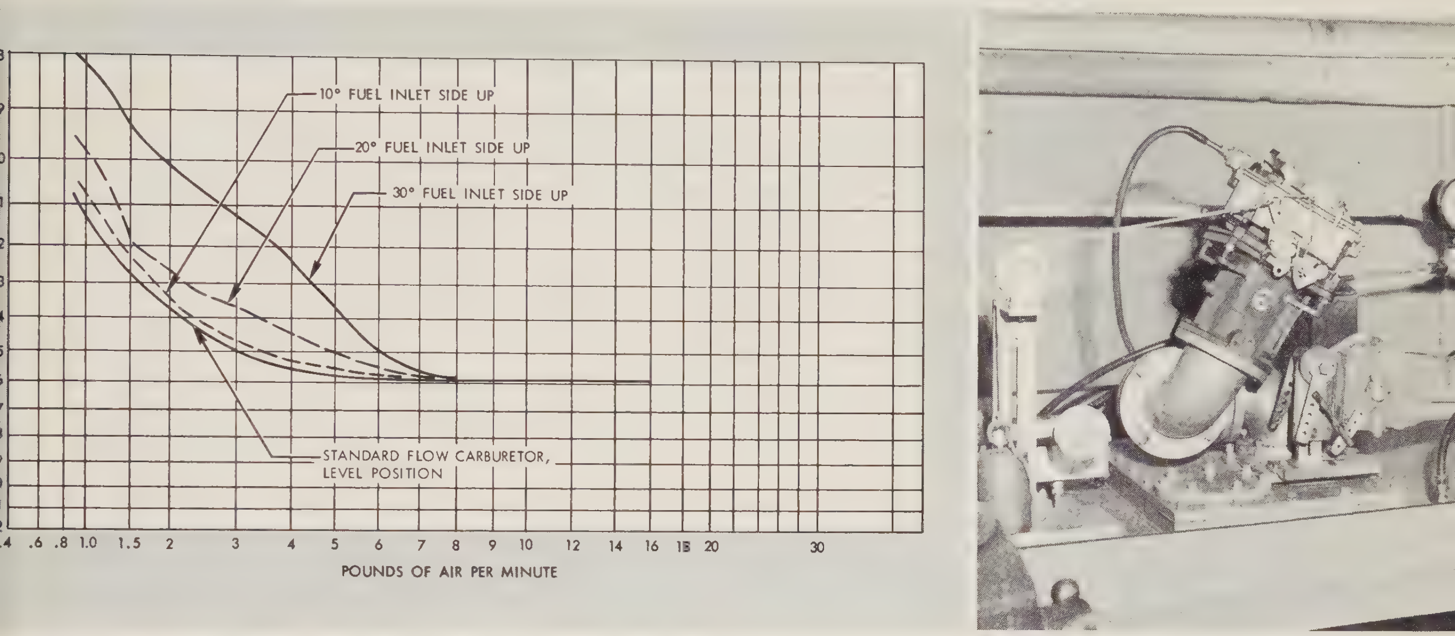


Fig. 9—This graph shows flow curves resulting from tests to simulate different angular positions of the vehicle such as on uphill or downhill driving. Known as *angularity curves*, they show the enriching effect caused by the car climbing various grades. (It should be noted that a 45° grade is equivalent to 100 per cent grade when expressed in grade percentages. Therefore, a 10° angle, for example, is equivalent to a 22¼ per cent road grade.) If the curve indicates excessive richness, the engineer must make design changes to compensate for driving performance that would be unacceptable to the car owner.

The photograph shows an angularity test arrangement where the carburetor is tipped with the fuel inlet upwards, thus simulating a steep uphill climb. Test set-ups also are made to simulate downhill operations as well as right or left turns.

- Capacity of air flow is limited only by capacity of the vacuum pump and not confined to a fixed volume as in a displacement box
- Remote throttle control becomes economically feasible because it is not necessary to open the tank to recharge the air cylinder
- Indicated air flow is read directly in pounds of air per minute
- Speed of operation is increased.

The usefulness of the flow box to the engineer's development work depends upon the accuracy of the data obtained. Consequently, an important laboratory procedure is the calibration of the air flow and fuel flow systems in the box. The general methods of calibrating the flow boxes at Rochester Products Division is described in the Appendix to this paper.

An example of a flow curve illustrates how it is used in an analysis of the flow characteristics of a typical automotive carburetor (Fig. 6). At an idle air rate of 0.9 lb of air flow per min, the flow curve shows that this carburetor delivers a rich mixture of 10.6 lb of air to one lb of fuel. By blocking off the main metering system of the carburetor, the idle system can be analyzed. The idle system flow to a rate of 7 lb of air per min is plotted on the graph. The *idle system* curve may be studied for optimum conditions by exploring the effects of varying idle tubes, idle channel restrictions, and idle air bleeds.

In a similar manner, by blocking the idle system the *main metering system* may be analyzed and further developed. In the case of the carburetor represented by this flow curve, the main metering system

Fuel enrichment is provided up to 19 lb of air per min as the throttle is opened. This portion of the curve is referred to as the *range of power valve* cut-in. As the valve is opened farther, this carburetor reaches its maximum air rate of 34 lb of air per min with a 13.8 to 1 air-fuel ratio.

If the vehicle were moving on a continually increasing grade with the throttle at wide open position, engine speed would gradually decrease causing a drop in manifold vacuum. To simulate this road condition in a flow box, the throttle is set at the wide open position and the vacuum applied is decreased. The data gathered from this technique produce a curve described as the *wide open throttle*

**Power valve* is a valve located within the fuel metering device which enriches the fuel mixture that operates when manifold vacuum drops to a predetermined value.

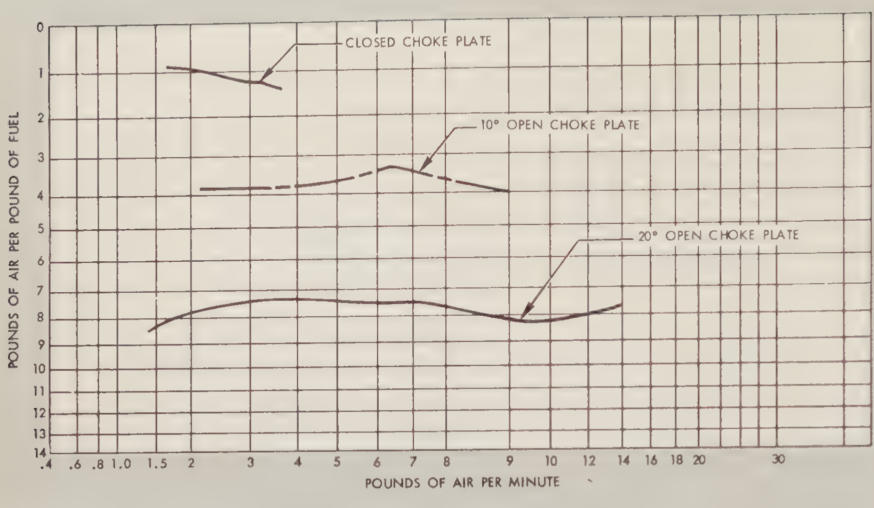


Fig. 10—Typical *choke curves* obtained in flow box tests are shown in this graph. These curves show the effect of fuel enrichment when the choke is locked in three selected positions: closed, 10° open, and 20° open. The top curve shows the effect of a closed choke plate as would be the case in initial cranking of a cold engine. The middle curve shows the effect of a choke plate in a 10° open position. This condition can occur during cranking and prior to first "fire." The bottom curve shows the effect of a choke plate in a 20° open position. This is the position the choke plate assumes immediately following first firing of a cold engine. The choke plate is moved to this position after first "fire" by a device known as the vacuum break. The vacuum break is composed of a piston actuated by manifold vacuum and connects through linkages to the choke plate. The engineer uses these curves as a guide for determining the amount of enrichment that will occur when starting a cold engine.

curve (Fig. 6). The shape and slope of this curve guide the engineer in developing maximum engine horsepower and torque.

The engineer also finds that in addition to development of the basic flow curve of the fuel metering device, the flow box may be used to prepare other types of curves which simulate conditions in different road tests. Examples of these curves are:

- *Crowd curves*†, made for a vehicle operating condition of constant uphill climbs at fixed throttle opening (Fig. 7), and for another condition of constantly increasing acceleration on a level road (constant manifold vacuum) (Fig. 8)
- *Angularity curves*, made for vehicle operating conditions under various angular positions such as uphill or downhill angles or side-to-side angles (Fig. 9)
- *Choke curves*, made to study the function of the choke in providing the proper enrichment to start a cold engine (Fig. 10).

†*Crowd curve* represents the type of curve produced when a given manifold vacuum value is held (crowded) while either varying throttle opening and/or increasing road load. Crowd curves also may represent a condition which exists when the manifold vacuum value is held constant while varying throttle position.

Summary

The flow curves described indicate how the flow box is used in carburetor development. If the crowd curves appear to be free of dips and humps, the carburetor will have no abnormalities in lean or rich conditions during the vehicle operation of constant uphill climb at fixed throttle opening. The crowd curve also provides information on carburetor performance just before and just after the power valve opens during a simulated vehicle operation of constantly increasing acceleration on a level road. At this operating condition, the design problem is to avoid too lean a mixture just before power valve opening (causing engine overheating) and to avoid too rich a mixture just after power valve opening (causing loss in economy). A study of the angularity curves and choke curves aid the designer in providing the proper metering for satisfactory vehicle performance.

Flow box tests are not a substitute for dynamometer tests or vehicular road tests. All three tests are used to measure the effectiveness of a fuel metering device. Because the flow box is an economical and versatile laboratory instrument, it plays an indispensable part in the development of fuel metering devices.

APPENDIX
Calibration and Correlation of Flow Boxes

Rochester Products Division uses nine flow boxes in connection with product development. Four other flow boxes are used in quality control work. Since there are minor structural and operational differences in the equipment of each flow box system, it is important that all measuring instruments be correlated to obtain uniform results. The following is a brief description of the procedure used to calibrate the air measuring and fuel measuring systems.

Air Flow Calibration

One displacement type flow box is designated as the master, or standard, box for all measurements. Three theoretically perfect venturi tubes are placed in the master flow box to calibrate the air flow. These venturii are of different sizes to provide a range of measurement from 0.2 lb to 45 lb of air per min. Varying pressure drops are imposed across them in increments of one-half in. of water varying from zero to 10 in. of water. The pressure differential is measured on a micro-manometer and the resultant rates are plotted against the pressure drop. Using these data and the three venturii, air flow rates are established on the other flow boxes. The inclined manometer scales on the flow boxes are then recalibrated insuring correlation with the master flow box.

Fuel Flow Calibration

Fuel flow calibration is based on measuring the weight of fuel flowing per unit of time. Accepted fuel flow calibration practice in the automotive industry has established that fuel having a specific gravity of 0.735 should be used as the standard. Therefore, it is necessary to calibrate instrumentation which measures the flow rate of this fuel in lb per min.

The equipment used is essentially a weight scale with a beam actuated timer switch which measures precisely the time required for a given weight of fuel to flow. With this information, the accurate rate of flow through the fuel measuring instruments on the flow box can be obtained. The fuel flow meter (used for high flow rates) is calibrated by adjusting its scale. The burette (used for low flow rates) is calibrated by applying an appropriate correction factor.

Evolution and Design of Compact, High Performance Pumps for Power Steering Systems

By ROBERT P. ROHDE
Saginaw Steering Gear Division

Meeting the demands of greater capacity, smaller size, reliability, and economy

Oil at the proper conditions of flow and pressure has a primary function in the power steering gear of an automobile. A significant component in the power steering system, therefore, is a reservoir and pump combination to supply oil at these conditions. As an automotive accessory the power steering pump must meet certain general requirements such as high efficiency, high performance, reliability, relatively light weight, and relatively low cost. These are objectives which engineers try to fulfill when designing almost any product. But the power steering pump must meet certain additional objectives which are unusual in nature and which present intriguing problems to the engineer. One such problem: the driven speed of the pump is not constant. Engineers at the Saginaw Steering Gear Division recently designed improved versions of a former pump to meet the requirements of 1959 and 1960 model year automobiles. This work included numerous examples of applying the principles which engineering students study in their machine design courses. For example, it involved considerations of different materials, mechanisms, and manufacturing methods. It involved the whole procedure of establishing and analyzing initial design objectives, applying fundamentals, sometimes accepting design compromises, and evaluating the resulting design.

The fundamentals and operating principles of steering an automobile with manual and power steering systems have been discussed in papers published in previous issues of the General Motors Engineering Journal[1-3]. This paper describes the progress in the design and development of the power source for power steering gears, namely the oil reservoir and pump. The other principal components of a hydraulic power steering system are the hydraulic valving means, actuated by steering wheel movements of the driver of the vehicle; the power cylinder, actuated by hydraulic fluid diverted to it by the valving means; and interconnecting piping.

Except for a portion of the first model year that power steering was used, Saginaw Steering Gear Division has always manufactured a vane type hydraulic pump to provide the power. The vane type pump was chosen because of its high efficiency at high pressure and high temperature, resulting from the self-wear compensating features of the vanes in contact with the internal cam surface (Fig. 1). Although the design progress of the pump has resulted in many improvements to the vane-cam relationship, the basic principle of the vane type is still in use.

Initially, the requirements of the pump were rather simple as the first power steering systems were merely assist systems to the conventional manual steering gears. However, as power steering has been called upon to do more and more work, the demands on the pump have multiplied many times. To illustrate this point, the first pumps had a flow requirement of 0.9 gpm and a minimum pressure relief setting of 750 psi. Comparing the hydraulic horsepower of these requirements to the modern requirements of 1.75 gpm with a minimum pressure relief setting of 1,100 psi, it is seen that the functional requirements of the pump are now 285 per cent of the original systems. This drastic increase in the functional requirements on the pump is explained by a number of influences:

- Heavier front end weights
- Wider tires
- Lower steering ratios
- Demand for lighter effort steering systems.

The requirements of a power steering system impose some very unusual conditions on the pump. Because of the speed variation of an automobile engine from idle speed to maximum speed in the ratio of approximately 12 to 1, and because the pump output must be adequate at idle speed, the pump displaces a very large excess of fluid at high speeds. This excess fluid is handled within the pump by a flow control valve. Handling this variation in speed and flow has resulted in some interesting problems in durability, noise, and operating temperatures.

Background of the Present Design

To better understand the design of the current type of pumps, it is of interest to review briefly the development of pumps used in previous car model years.

There were no major changes made to the pump through the model years 1953-54-55 (Fig. 2). During this period, however, many design refinements in suction-discharge timing and in component clearances were made to improve the efficiency and to reduce noise. Other changes were concerned with reducing the cost of the reservoir such as using lighter gage material, reducing the height, and using a die cast reservoir cover instead of a fabricated steel cover.

Although Buick, Oldsmobile, Pontiac, and Cadillac automobiles have always used a pulley driven pump, a new, direct driven pump, located at the rear of the generator, was manufactured for Chevrolet automobiles beginning with the model year 1955 (Fig. 3). At this time, the maximum generator speed was 9,000 rpm and the flow-pressure requirements to steer a Chevrolet were relatively low. Attempts to provide a generator pump with increased flow-pressure characteristics for the heavier cars, Buick, Oldsmobile, Pontiac, Cadillac, seemed to be incompatible with the high speed operation of a high performance hydraulic pump. Although the generator driven pump originally offered Chevrolet some outstanding advantages, such as elimination of an extra drive belt and reduced size and weight, these advantages have gradually been offset by the development of the inherent advantages of operating a hydraulic pump at lower than generator speeds.

Steering Advances Increased Requirements on Pump

In 1956, with the advent of the in-line steering gear, the pump requirements were drastically increased to require a minimum pump flow of 1.50 gpm with

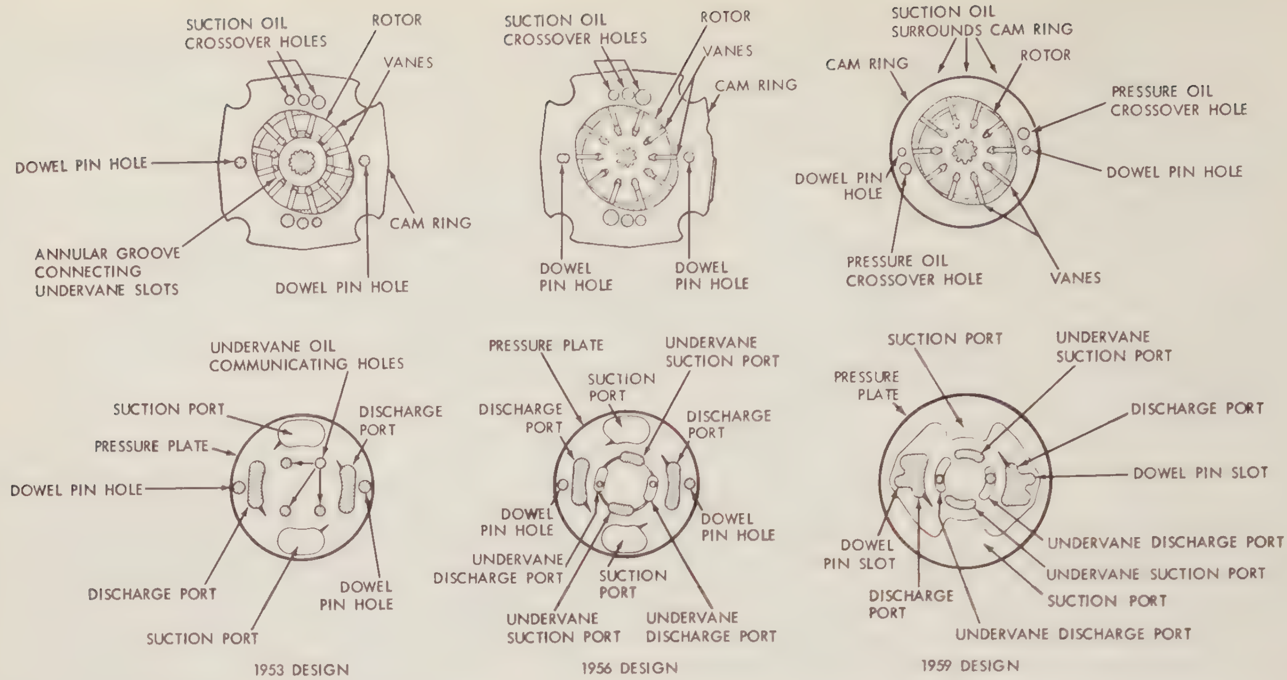


Fig. 1—The vane type, double acting pump has been the basic design of powering steering pumps built by Saginaw Steering Gear Division as shown in this comparison (from left to right) of three pumps. In all cases the pumping means consists of (a) a *cam ring* that has two rise and two fall areas resulting in a hydraulically balanced system, (b) a rotor with vane slots, (c) vanes slidably mounted in the rotor slots, and (d) porting means in a pressure plate. The vanes are held in intimate contact with the cam ring by centrifugal force and hydraulic pressure applied to their undersides. The upper drawings are plan views of the rotor, vanes, and cam ring assembly. The lower drawings show the matching pressure plate for each design. Assembly views of the complete pumps using these designs are shown in Figs. 2, 4, and 6.

At the left is the 1953 model, 12-vane rotor with an annular groove connecting all undervane slots. This annular groove was openly connected to the high pressure side of the pump through the four holes toward the center of the pressure plate. The cam shape was based on simple harmonic motion.

In the center is the 1956 model using a 10-vane rotor which provided greater rotor segment strength as well as eliminating two vanes per pump. To improve durability, the undervane annular groove was replaced by the four annular segments toward the center of the pressure plate. The undervane suction ports were openly connected to the high pressure chamber. The undervane discharge ports also were connected to the high pressure chamber but through a restricting orifice, the purpose of which was to insure intimate contact with the cam ring as the vanes retract. Previous experience showed that the vanes had a tendency to skip away from the cam in this area. The cam shape was based on uniform acceleration (parabolic) which also improved durability.

The 1959 design at the right incorporates the features of the 1956 design but the pump ring is completely surrounded by oil in the pump housing, thereby minimizing suction filling problems.

pressure reliefs as high as 1,000 psi. To meet these requirements, the pulley driven, vane type pump design was retained but substantial improvements were made. This pump resembled the previous models, but design changes were made in the vanes and cam ring to increase capacity. Othee components were changed to reduce cavitation, noise, and wear (Fig. 4).

Model years 1957 and 1958 saw the continued usage of the basic 1956 pump design, although some design changes were made through these two model years to reduce the cost of the pump. These changes were primarily in the area of the shaft bearing design, gradually evolving to the use of babbitt bushings in 1958.

It should be pointed out here that the inherent advantages of operating the pump at pulley driven speeds had not yet been exploited sufficiently to overcome the advantages of the generator driven pump for the lower flow-pressure requirements of Chevrolet cars.

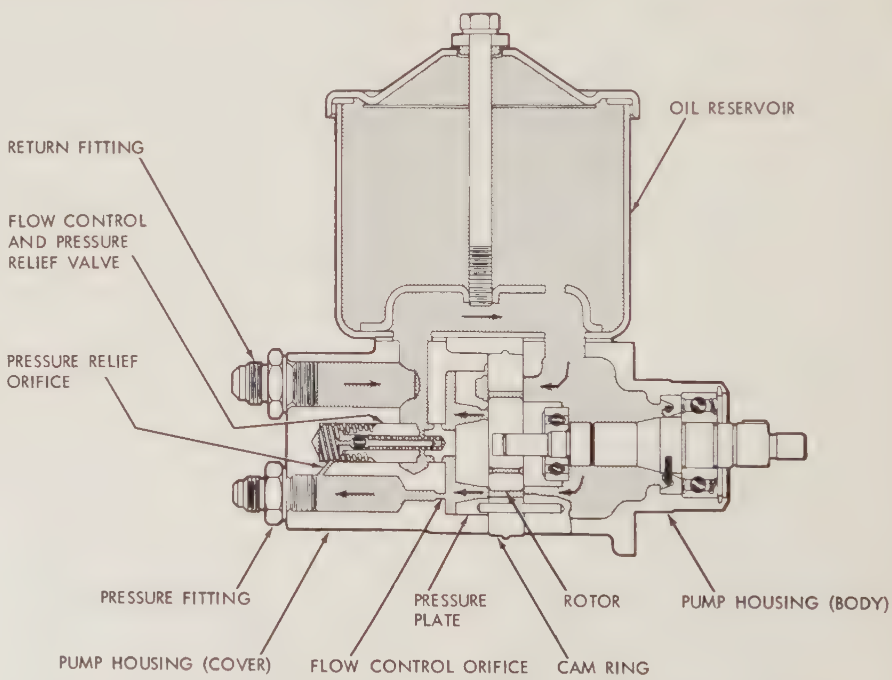


Fig. 2—This assembly view shows the vane type pump manufactured by Saginaw Steering Gear from 1953 through 1955. During these years, design and test time was devoted primarily to improving and refining the functional characteristics of the pump. Some cost reduction changes were made to the reservoir and reservoir cover.

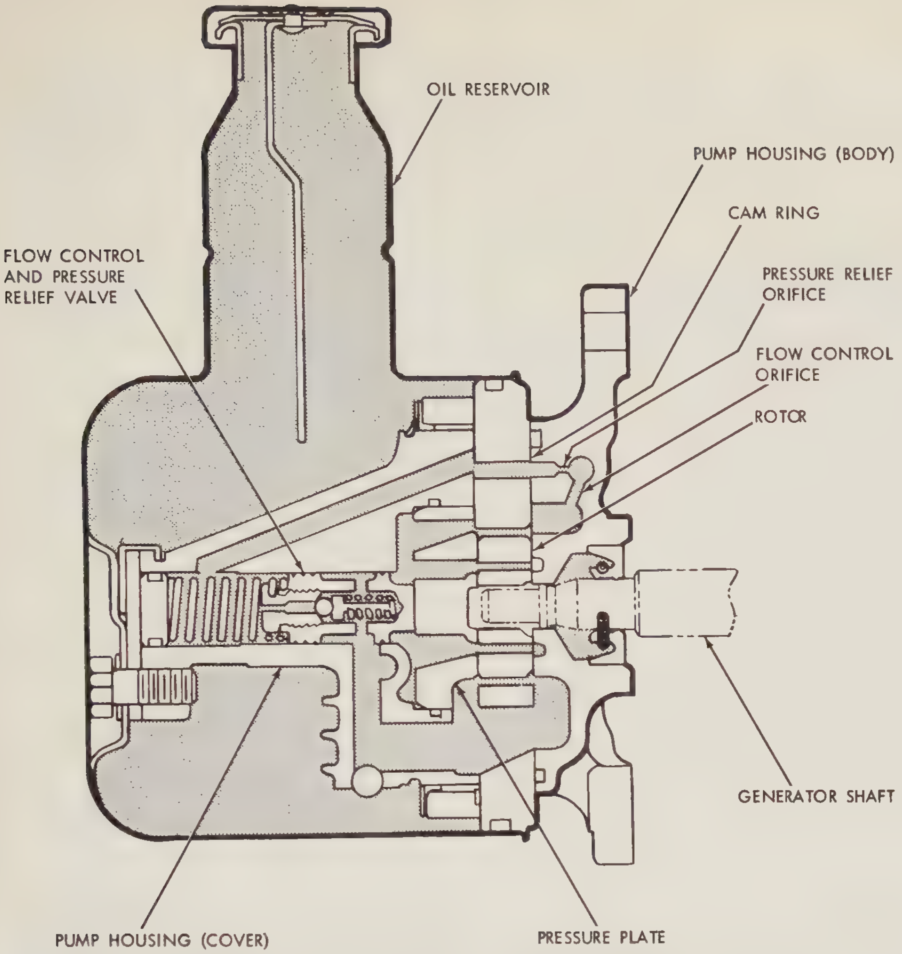
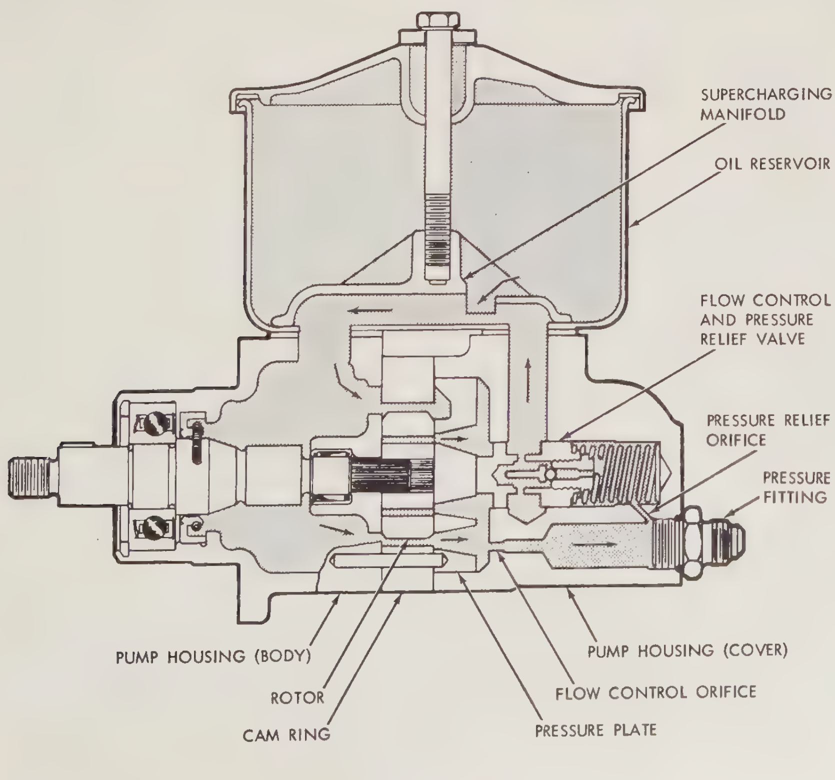


Fig. 3—Chevrolet automobiles during the 1955 through 1959 model years used the direct driven, vane type pump shown here. This pump was driven by the generator shaft extending into the pump rotor. It operated at twice the engine speed. Pulley driven pumps operate at or near engine speed; pump speed to engine speed ratios are in the range of 1 to 1 and 1.25 to 1. The development of the inherent advantages of running a high performance hydraulic pump at engine speeds gradually overcame the advantages of the generator driven pump.



Advances in Steering Gears and Assembly Methods Led to Redesign

With the advent of the Saginaw rotary valve steering gear for 1959 model cars, which increased requirements on the pump, and with the trend toward continuous flow assembly methods, it was evident that a complete redesign of the pump was desirable.

The objectives established as design criteria for this new pump were as follows:

- Provide a smaller pump to diminish installation problems caused by less and less available space through styling changes
- Provide a pump enclosed in a reservoir to minimize the possible external leakage points
- Provide a lower cost pump
- Provide a one-piece housing suitable for continuous flow machining and assembly
- Provide a more efficient intake supercharge system to lessen noise and improve durability, by reducing cavitation
- Provide an intake system surrounding the cam ring to further reduce cavitation
- Provide a flow control system to allow external pump flow to drop off at higher speeds, thereby decreasing back pressure through the steering gear with a resulting reduction in temperature
- Provide a higher capacity pump capable of handling the increased requirements of the rotary valve steering gear

Fig. 4—An improved vane type pump was manufactured from 1956 through 1958. Although similar in appearance to the previous model (Fig. 2), there were a substantial number of design changes. The inner shaft bearing was changed from a ball bearing to a less costly needle bearing. The pump capacity was increased from 1.0 gpm at idle speed to 1.50 gpm by increasing the cam ring thickness and by reducing the number of vanes from 12 to 10. Suction and discharge timing were revised to provide more quiet operation. An inlet supercharge system was used. This system utilized the oil by-passed internally in the pump by the flow control valve to provide positive suction pressure at high speeds. This supercharging, which reduced cavitation, noise, and wear, was accomplished in the supercharging manifold. Oil returning from the steering system was directed back to the reservoir rather than to the suction side of the pump. Easier air bleeding and cooler operation resulted from this change. Undervane timing (Fig. 1) was incorporated to improve durability. Joints where external leakage could occur were revised to conform with up-to-date principles of seal design and application.

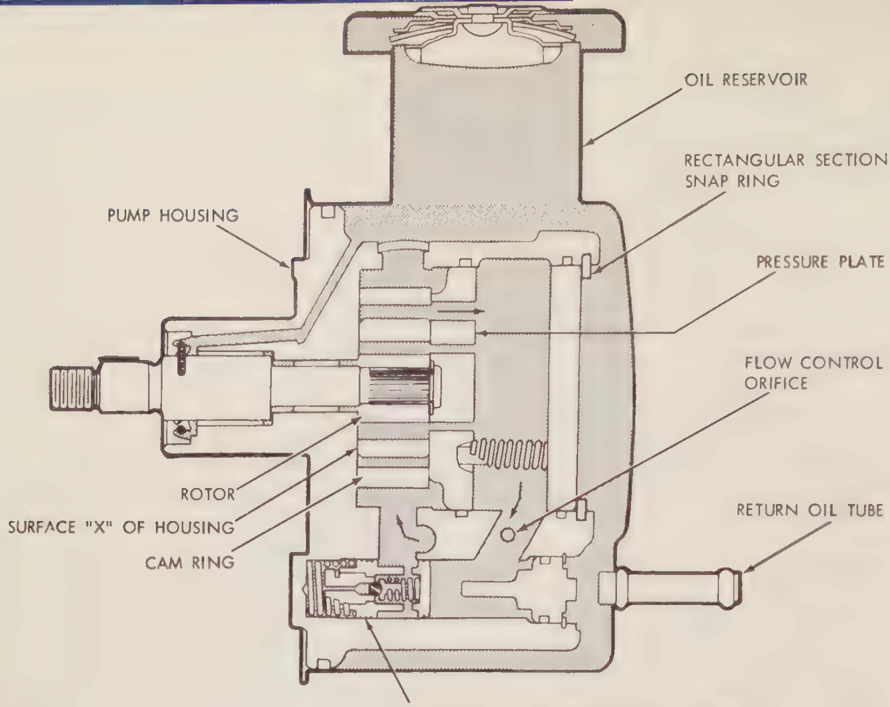


Fig. 5—This pump was the initial version built to meet the design objectives of the 1959, pulley driven, high performance pump. The one-piece housing permitted most of the machining and assembling to be done from one end. Compared to previous model pulley driven pumps (Figs. 2 and 4), this pump was considerably more compact which meant fewer installation problems. The number of high pressure leakage points was reduced to one by surrounding the pump with its reservoir. A flow control system (explained in Fig. 6) that allowed the external pump flow to drop off at higher speeds was incorporated in the design. This pump was capable of delivering 1.75 gpm at engine idle with pressure reliefs as high as 1,300 psi. After evaluation of this design, an improved version was developed which was used in 1959 cars.

- Develop an application of powdered metal parts for the side plates of the cam ring and rotor to better control the suction and discharge timing, with resulting noise reduction
- Improve pump efficiency.

Experience indicated that, from a manufacturing point of view, it was essential that the two-piece housings of previous pump designs be replaced with a one-piece housing. Also, it was desirable that most of the machining be done from one end of the housing, and that the design be suitable for assembly from one end so that the pump, once on the assembly line, could be continuously assembled without removal from the line. It was obvious that the flow control valve could no longer be in line with the shaft and rotating group, but would have to be offset to one side of the pump housing.

Evaluating the Design and Solving the Remaining Problems

The initial design (Fig. 5) met most of the established objectives. However, after a thorough evaluation of the design,

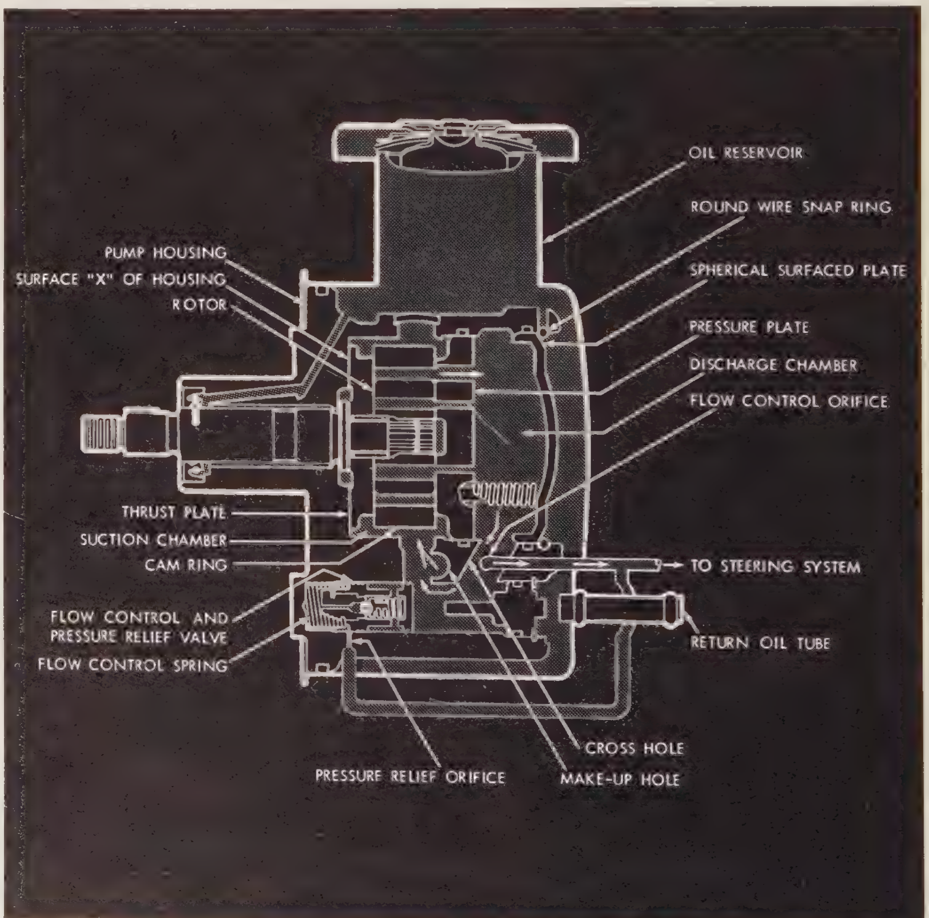
Fig. 6—The final design of the 1959 pump successfully incorporated means to supercharge the pump inlet efficiently and to cause the external pump flow to drop off at high speeds.

All fluid displaced by the vanes is discharged through the pressure plate into the discharge chamber. From the discharge chamber, the oil flows into the cross hole which is controlled in size to provide definitely known oil velocity. From the cross hole a certain quantity of oil passes through the flow control orifice and then to the steering system. Notice that the downstream side of the flow control orifice is connected to the spring side of the flow control valve. When the quantity of oil displaced exceeds the predetermined system requirements, the pressure drop through the flow control orifice exceeds the force of the flow control spring and the flow control valve will move back toward the spring, thereby providing flow control as shown.

Supercharging occurs as a result of the oil under pressure discharging from the flow control valve at high velocity, picking up the make-up oil from the reservoir through the make-up hole. Then by a gradual reduction of velocity in the passages downstream from the flow control valve, the velocity energy is converted into supercharge pressure in the suction chamber.

The pressure relief pilot valve is contained inside the flow control valve. When pump pressure exceeds the pre-determined pressure, the pressure relief ball opens, allowing a small amount of oil to flow from the downstream side of the flow control orifice through the pressure relief orifice and through the flow control valve and into the suction chamber. This flow of oil causes a pressure drop across the pressure relief orifice, thus creating a pressure unbalance which moves the flow control valve back against the flow control spring and allows the major portion of oil to bypass in the same manner as is accomplished by flow control.

Drooping flow control occurs as follows: as the quantity of displaced oil increases, the velocity in the cross hole increases creating a venturi effect on the flow control orifice which actually reduces the amount of flow to the steering system. The effects are shown graphically in Fig. 8.



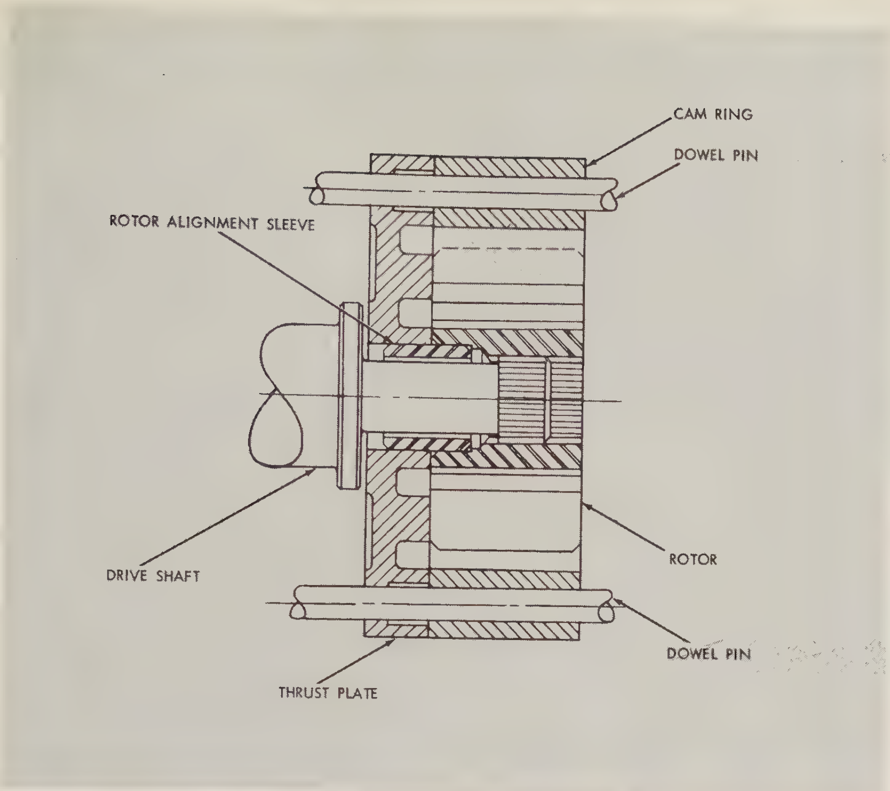


Fig. 7—Concentric rotation of the rotor with respect to the cam ring, essential for noise-free operation with the reduced bearing span of the 1959 compact pump, is assured with the use of the rotor alignment sleeve. This sleeve is press fitted into the rotor and runs with a bushing fit in the central hole of the thrust plate. Concentric rotation is thereby assured, as the thrust plate and cam ring are doweled together.

based on laboratory tests and meetings with manufacturing, tooling, and process engineering personnel, it was apparent that the following problems existed:

- The use of two bushings on the drive shaft was undesirable from a machining and assembly standpoint
- The reduced span of the bushings, necessary to shorten the pump, resulted in a slight eccentric rotation of the drive shaft with respect to the internal cam surface, causing noise
- The desired reduction of cavitation did not materialize
- No provisions were made to improve mechanical efficiency
- The design of the retaining ring holding the pump end plate caused failures in the pump housing. Because of the large area of the end plate which received full pump pressure and, therefore, relatively large forces, the rectangular section retaining ring caused failures in the housing due to stress concentration at the base of the ring groove. Another disadvantage of this retaining ring design was that, in this large size, it was relatively expensive and difficult to assemble
- The powdered metal pressure plate scored too easily and deflected excessively under pressure
- The short span bushings failed repeatedly on laboratory tests even with shaft surface finishes as low as five micro-inches
- Failures of the larger displacement cam pointed to the need for improvements in design and metallurgy
- It was determined that it was not possible to machine the bottom surface (surface *X*, Fig. 5) of the housing flat enough to operate against the rotor and vanes.

Solving these problems required further studies which ultimately resulted in the final design of the high performance pump for 1959 model automobiles (Fig. 6). For example, in the final design a single bushing was used to replace the two used on the initial design. Because of space limitation, it was not possible to lengthen the bushing span to solve the noise and bushing durability problem. The solution to the durability problem was rather simple, once uncovered. It was determined by test that the problem

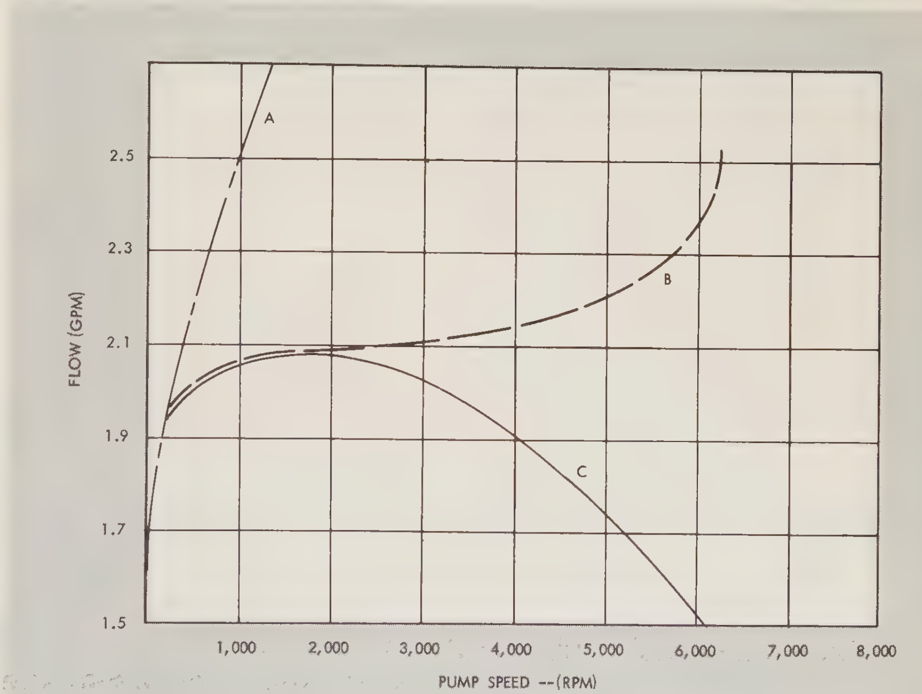


Fig. 8—How the output of typical power steering pumps varies when speed is increased is shown by these curves. Curve A shows the output flow for a pump that does not have a flow control valve. Curve B shows output for a typical flow controlled pump such as the 1956 model (Fig. 4). Curve C shows the drooping flow characteristic of the 1959 and 1960 pumps. This drop off of flow at high speeds results in reduced back pressure through the steering system with resulting reduction in operating temperature.

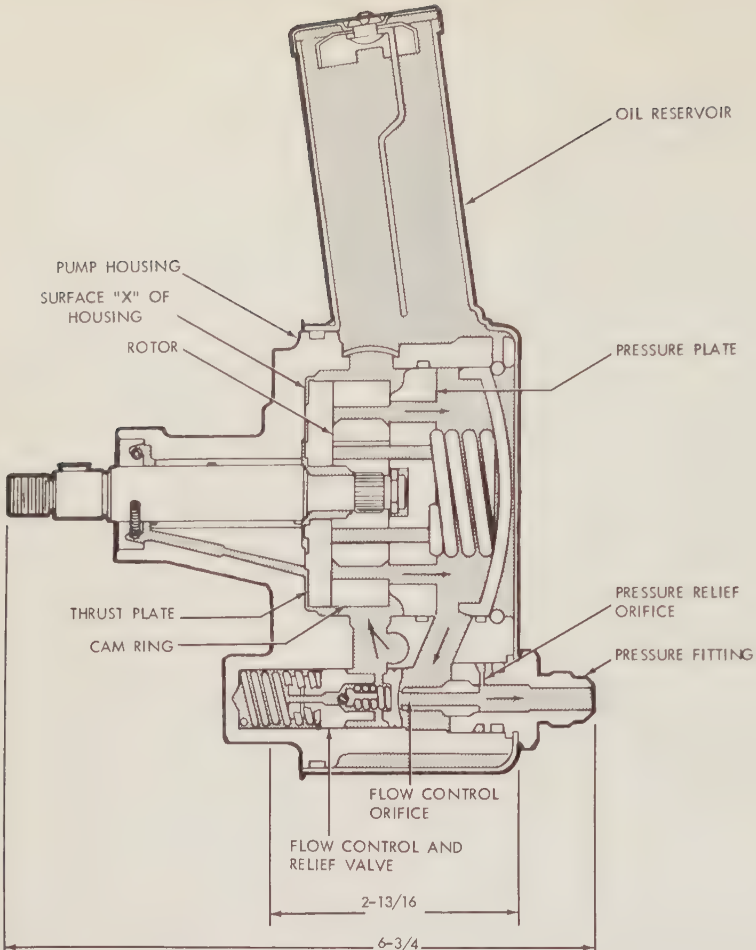


Fig. 9—The final design of the 1960 Chevrolet pump, although slightly different in the arrangement of some parts, functions in exactly the same manner as the 1959 pump designed for use in the other GM cars (Fig. 6). A comparison of pump sizes shows how this design satisfied the objective of compact size for mounting between the engine and fan. The overall length of this pump is 6¾ in. and the distance from the mounting surface to the rear of the body is 2¹³⁄₁₆ in. In the case of the 1956 design, for example, these corresponding dimensions are 9¾ in. and 5¹³⁄₁₆ in. To meet another objective of the design, the 1960 pump can accommodate larger size cam rings and therefore increase its output should this be required in future pump models.

did not lie so much in the surface finish of the shaft, but in the fact that even a ground surface has relatively sharp edges. This condition caused damage to the soft babbitt bushing. It was found that by roll-burnishing the shaft, the durability of the bushing was satisfactory even though the burnishing did not improve the surface finish value, as such.

The pump noise problem created by the short bushing span was solved with the use of a rotor alignment sleeve (Fig. 7). This sleeve, press fitted into the rotor and running with a bearing fit in the thrust plate, insured concentric rotation of the rotor with respect to the internal cam surface, independent of the condition of rotation of the shaft. It was necessary to make a looser fit between the pump rotor and drive shaft to insure against a binding condition with the alignment sleeve.

The desired reduction of cavitation was accomplished by providing, as nearly as possible within the limitations of reasonable manufacturing practice, a gradual increase in passage area as the flow of oil progressed from the flow control valve to the suction portion of the cam ring. This gradual increase in area allowed the efficient conversion of the kinetic energy of the oil at the flow control valve to static pressure at the suction ports. The static pressure provided a supercharge condition at the suction ports, thereby reducing cavitation to an acceptable point (Fig. 6).

Mechanical efficiency could have been improved slightly by replacing the babbitt bushings with ball bearings, but this would have increased the overall cost. Instead, higher efficiency was obtained by reducing the internal pressure losses of the pump. Oil was discharged from both sides of the pump ring instead of only from the pressure plate side. This double discharge drastically reduced the pressure loss of the oil flowing from the cam into the discharge chamber of the pump.

The housing end plate was redesigned to allow the use of a round wire snap ring (Fig. 6). The plate was made dome-shaped to strengthen the plate itself and to provide an angular contact with the snap ring so that increasing loads on the end plate forced the snap ring still tighter into its groove, thereby providing a self-locking ring. This design was satisfactory from a strength standpoint and also was less costly than the original design.

An improvement in durability was obtained by specifying an alloy iron casting for the cam ring. This alloy iron had to be carefully controlled to provide desirable graphite flake types, distribution, and size as well as eliminating excess free ferrite and carbides. This iron also had to be hardenable to provide a final hardness of Rockwell C-50.

The scoring of the powdered metal pressure plate which occurred in tests of the previous design was eliminated by applying a steam oxide treatment before machining the plate and a phosphate type anti-friction coating treatment after machining. The steam oxide treatment served two purposes: (a) by closing the pores of the powdered metal it prevented the influx of acid from the phosphate type anti-friction coating treatment, and (b) the modulus of elasticity was increased sufficiently to prevent excessive deflection.

The problem of machining a housing surface that would be sufficiently flat for operation against the pump rotor and vanes was solved by using a separate thrust plate between the housing (surface *X*, Fig. 6) and the rotor-vane-ring assembly. The final design incorporated a powdered metal thrust plate, machined flat, and assembled into the pump.

The excess fluid produced by the pump at high rotating speeds was handled by a flow control valve. This flow control valve also incorporated means to provide a pressure relief valve for the system (Fig. 6). A drooping flow characteristic at high speeds was obtained in this design by a venturi effect on the flow control orifice leading to the steering gear supply hose (Fig. 8).

After making these design changes, final evaluation and testing showed that the design met the objectives satisfactorily.

This pulley-driven pump was used in power steering systems for 1959 and 1960 model Cadillac, Buick, Oldsmobile and Pontiac cars.

Pump for Chevrolet Cars Had Different Requirements

The power steering system for 1959 Chevrolet cars continued to use the generator mounted, direct driven pump. However, in the development of the 1960 model Chevrolet, it appeared that the improved performance of the pulley driven pump would make this design outweigh the advantages of the direct driven pump. Thus, the pulley driven design was planned for the 1960 Chevrolet. While it seemed that the task of adapting this pump to the Chevrolet would be a relatively simple one, it was complicated by the lack of available space for locating a pump of this size. As a result, a redesign of the pump had to be made to mount the pump between the engine and the fan. This location offered the advantages of reduced temperature and lower noise level compared to the generator mounting.

In making this redesign, the following objectives were established:

- Incorporate all desirable features of the 1959 pulley driven pump
- Reduce length of pump sufficiently to mount in front of the engine
- Reduce operating temperature
- Reduce noise level
- Design one pump that could be used without modification on all three Chevrolet engines: the *W* (348 cu in. V-8), 283 cu in. V-8, and the 6-cylinder engine
- Although the pump could be designed to meet only the present flow requirements for Chevrolet, it should be designed to allow incorporation of a larger displacement cam in the event that this pump should be used with higher flow requirements at a later date.

The objectives which applied to the larger, 1959 pulley driven pump were met by following the same principles that led to the successful design of that pump. The basic design is, in fact, the same (Fig. 9). However, careful study and exhaustive testing showed that wall thickness and oil passages could be reduced to meet the space requirements. Even after this test program, two problems still existed.

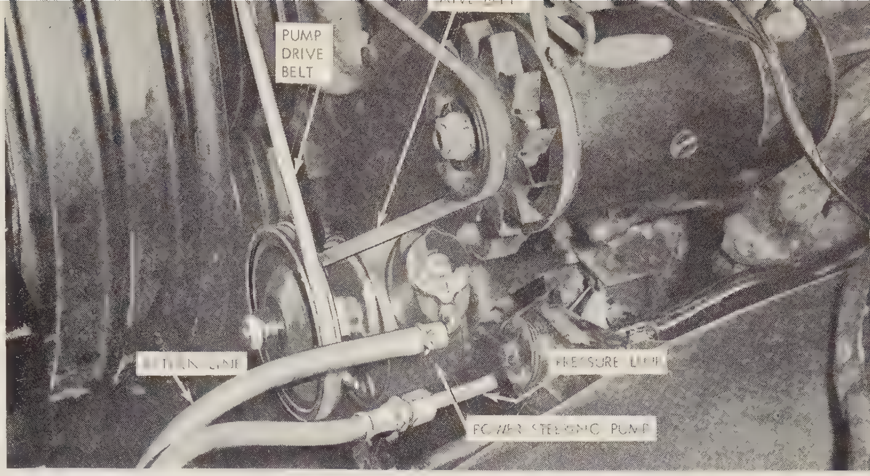


Fig. 10—The location of the 1960 Chevrolet power steering pump is shown in this photograph. The pump body fits between the path of the generator belt and the front of the engine.

Based on experience with the 1959 pump, it was known that it was not possible to machine the bottom surface (surface *X*, Fig. 9) of the housing perfectly flat. To insure against seizure problems caused by convexity of this surface, all machining was done to hold this surface flat, with all tolerance going toward concavity of the surface. The concavity of the surface plus the deflection under pressure of the relatively thin wall section of this face (necessitated by space limitations) allowed the center portion of the thrust plate to deflect away from the rotor, thereby allowing excessive internal leakage in the pump.

The second problem was an inadequate capacity of the reservoir to take care of the thermal expansion and contraction of the oil.

The application of stress-coat techniques led to the solution of the deflection problem of the bottom surface of the pump housing. It was not possible to add material to the entire front face of the housing because the generator drive belt crossed in front of a portion of this surface (Fig. 10). The use of stress-coat showed where the front face could be strengthened sufficiently to limit deflection and still provide adequate clearance for the generator drive belt.

An elliptical filler tube brazed on the reservoir body at a compound angle satisfied the condition of providing one pump to fit three engine models. This filler tube arrangement provided adequate reservoir capacity within the space and clearance limitations of the three engine models.

Further testing verified that the final design of this pump for the 1960 Chevrolet met the remaining objectives such as lower operating temperatures and lower noise level.

Summary

In recent years the functional demands on pumps for power steering systems have increased substantially while the available space for such pumps has gradually decreased. To keep pace with these changing requirements, Saginaw Steering Gear Division engineers have developed compact, high performance pumps which can be manufactured economically by the latest type of processing equipment.

These improved designs began by establishing certain design objectives. On the basis of the objectives and making the best possible use of previous experience, experimental pumps were constructed. Testing of these pumps uncovered some remaining problems. After a thorough study of these problems, final designs were possible which satisfied the original design objectives, and were accepted for use on 1959 and 1960 General Motors automobiles.

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1. Lincoln, C. W., "A Summary of Major Developments in the Steering Mechanisms of American Automobiles," *General Motors Engineering Journal*, Vol. 2, No. 2 (March-April 1955), p. 2-7.
2. Marquis, Donald P., "How Fundamentals Are Applied to the Design of Safe, Efficient Automotive Steering Systems," *General Motors Engineering Journal*, Vol. 4, No. 4 (October-November-December 1957), p. 2-7.
3. Thompson, Blair, "Improving Product Design: Development of the Rotary Valve Power Steering Gear," *General Motors Engineering Journal*, Vol. 6, No. 2 (April-May-June 1959), p. 7-13.

Facts Engineers and Inventors Should Know About the United States Patent Office

By GEORGE L. DeMOTT*
Patent Section
Washington Office

What the patent application contains and how it is handled in the patent office

INVENTION may be mothered by necessity but without a doubt it is cherished, sustained, and promoted by the United States Patent Office. In lieu of a birth certificate given to an inventor for his "baby," the Patent Office issues a patent which contains a grant to the inventor of "the right to exclude others from making, using, or selling the invention throughout the United States" for a period of 17 years.

This article briefly describes the organization and operation of the United States Patent Office located in the Department of Commerce building in Washington, D.C., and briefly sets forth the steps through which a patent application must proceed before it matures into a patent.

How the United States Patent Office is Organized

The chief function of the Patent Office is to administer the patent laws as they relate to the granting of patents. The examination of applications for patents is the most important function of the Patent Office. For this purpose 74 examining divisions have been set up, each division having jurisdiction over certain fields of invention. Each division is headed by a Primary Examiner and is staffed by a number of Assistant Examiners. The Examiners perform the work of examining applications for patents and determine in the first instance whether patents can be granted.

In addition to the examining divisions, the Patent Office has several sections, or branches, which perform various services such as receiving and distributing mail, receiving new applications, inspecting drawings, and recording assignments.

Within the Patent Office there is a Board of Patent Appeals which consists of the Commissioner of Patents, three Assistant Commissioners, and 15 Examiners-In-Chief. It is to this Board that an inventor may appeal if the Examiner handling his application holds that what the inventor has done does not amount to invention. In most cases, a panel of three members of the Board of Appeals is present at a hearing and decides the question on appeal.

In addition to the Board of Appeals, there is the Board of Interferences whose function it is to determine which of two or more inventors is entitled to a patent in the event they are claiming substantially the same patentable invention. Each party to an interference proceeding must submit evidence of the facts proving when he made the invention and, upon review of these facts, the Board of Interference Examiners will determine to whom a patent may be granted.

At present there are about 2,200 employes in the United States Patent Office of whom about one-half are Examiners. These Examiners have technical and legal training. Patent applications are received at the rate of about 70,000 per year and of these about 40,000 mature into patents. Approximately one per cent of the applications filed become involved in an interference proceeding.

Scientific Library and Search Room

The Patent Office has a scientific library containing over 70,000 volumes of scientific and technical books, about 60,000 volumes of periodicals, and copies of over 7,000,000 foreign patents.

In addition to the scientific library, the Patent Office maintains a search room for the benefit of the public in searching and examining U. S. patents, and within or about this room is a set of U. S. patents granted since 1836. The number of U. S. patents now is approaching 3 million; patent No. 2,900,000 was issued on August 18, 1959.

For purposes of searching, the Patent Office has classified patents into approximately 307 classes and over 52,000 subclasses. It is through these classes and subclasses that one searches to determine if his invention is in fact different from anything previously patented.

Examination of Applications in the Patent Office

According to statute, only the inventor may apply for a patent with certain exceptions, and his application for patent must include the following:

(a) A specification describing the invention and concluding with one or more claims which particularly define the invention

(b) A drawing in those cases where a drawing is possible

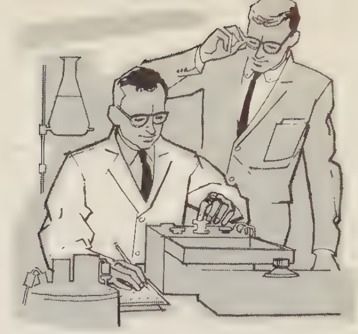
(c) A written document which comprises a petition to the Commissioner of Patents and an oath that the applicant is in fact the inventor of the described invention

(d) A filing fee of $30 plus an additional $1 for each claim in excess of 20.

Upon being filed in the Patent Office and accepted as a complete application, the application is assigned to one of the 74 divisions and to one of the 1,200 Examiners for examination. After studying the application and reviewing the patents and other technical literature available to him, the Examiner will either indicate that the application may become a patent or he will reject the application. The Examiner will put his decision in writing and this decision is then known as an *Action.* It is mailed to the attorney or agent of the applicant, or to the applicant himself if he has no attorney. In his Action, the Examiner will use such terms as "anticipation," "fully met," or "lack of novelty," if the invention as claimed is not new in the Examiner's opinion. If there is some difference or some novelty, but the difference over what is old is not considered to justify a patent, the Examiner may use such terms as "unpatentable over . . ." or "lacking invention over . . ."

*Deceased 1959

Notes About Inventions and Inventors



Contributed by

Patent Section

Detroit Office

THE following is a general listing of patents granted in the names of General Motors employes during the period July 1, 1959, to September 30, 1959.

AC Spark Plug Division
Flint, Michigan

• **Werner F. Schultz,** (*B.E.E., University of Detroit, 1938*) development engineer, inventor in patent 2,893,233 for a cap device for a filler pipe.

• **Alfred Candelise,** (*Ph.D. degree in electrical-mechanical engineering, University of Naples, 1923*) staff engineer, inventor in patent 2,894,315 for a spark plug and method of making same.

• **John D. McMichael,** (*Michigan State University*) senior project engineer, inventor in patent 2,905,268 for a cleaner silencer assembly, and in patent 2,904,129 for a low pass filter type cleaner silencer unit.

• **Homer R. Hastings,** (*B.S.E.E., Michigan State University, 1939*) staff engineer, and **Bryce L. Stevens,** (*B.S.M.E., University of Michigan, 1951*) project engineer, inventors in patent 2,899,517 for liquid level signal arrangements.

• **Argyle G. Lautzenhiser,** (*B.S.E.E., Tri-State College, 1940*) senior project engineer, and **Kenneth E. Faiver,** (*B.S. E.E., Notre Dame University, 1924 and Ph.D., Rensselaer Polytechnic Institute, 1927*) senior project engineer, Oldsmobile Division, inventors in patent 2,901,241 for a control device.

• **William C. Cole,** (*Michigan College of Mining and Technology*) section manager, aircraft spark plug engineering, inventor in patent 2,902,617 for a spark plug and method for making same.

• **Roscoe M. Wheeler,** automotive engineer, inventor in patent 2,902,683 for signal systems with warning devices.

***Inventors' names marked with an asterisk have biographical listings noted previously in this issue's Notes About Inventions and Inventors.**

• **Joseph Zubaty,** (*M.S.M.E., University of Prague, 1918*) staff engineer, inventor in patent 2,904,027 for a fuel supply system.

• **Wesley W. McMullen,** (*B.S.M.E., University of Michigan, 1934*) staff engineer, inventor in patent 2,905,265 for a cleaner silencer unit.

• **John D. McMichael*,** and **Wesley W. McMullen*,** inventors in patent 2,906,370 for an air cleaner and silencer assembly.

• **Arthur V. Somers,** (*B.S. in chemistry, Michigan State University*) senior experimental chemist, and **Karl Schwartzwalder,** (*B.Cer.E., 1930, and M.S., 1931, The Ohio State University*) director of research, inventors in patent 2,906,909 for cement for spark plugs and the like.

After the Action has been mailed, the applicant must reply within six months. If the Action consists of a Notice of Allowance—which rarely happens—the applicant must pay the final fee of $30. On the other hand, if the Action rejects some or all of the claims, he may amend them in view of the art cited by the Examiner and ask for re-examination or reconsideration, and he must distinctly point out the supposed errors in the Examiner's Action.

On the second or any subsequent examination by the Examiner, if the case is still not considered allowable, a final Action will be sent out by the Examiner to which the applicant's response is, for all practical purposes, limited to an appeal to the Board of Appeals. In making such a final rejection the Examiner repeats or states all grounds of rejection considered applicable to the claims in the case, and it is upon these grounds that the applicant may request a review by the Board of Appeals. The Board of Appeals may reverse the Examiner and direct him to allow the claims on appeal, or they may affirm the decision of the Examiner. If the Examiner's decision is affirmed, the applicant may further appeal on the record to the Court of Customs and Patent Appeals or may bring a civil action against the Commissioner of Patents in the District Court for the District of Columbia requesting the Court to find that the applicant is entitled to receive a patent for his invention.

If on re-examination or reconsideration of the application the Examiner finds that the application is allowable, the applicant is sent a Notice of Allowance indicating that a patent will be issued provided that a final fee of $30 and $1 for each claim in excess of 20 is paid within six months from the date of the Notice of Allowance. When the final fee is paid the patent ordinarily issues within five to seven weeks after the date of payment. The inventor is then sent his grant and copies of the patent are printed and made available to the public.

Allison Division
Indianapolis, Indiana

• **Dean K. Hanink,** (*B.S.Met.E., University of Michigan, 1942*) chief metallurgist, inventor in patent 2,893,349 for an apparatus for removing excess coating from a poppet valve.

• **William S. Castle,** (*B.S. University of Illinois, 1942*) group leader on turbines and compressors, Jet Engine Design Engineering Department; **David E. Schnable,** (*B.S.M.E., Tri-State College, 1939*) senior project engineer; and **Clair A. Short, Jr.,** (*M.S.M.E., Georgia Institute of Technology, 1936*) assistant chief design engineer, fuels and controls systems, inventors in patent 2,893,353 for a three-position actuator cylinder.

• **Charles J. McDowall,** (*B.S.M.E., University of Florida, 1927*) technical assistant to the director of engineering; **Victor W. Peterson,** (*B.S.M.E., Rose Polytechnic Institute, 1939*) engineer, Advance Design and Development Engineering Department; and Wallace Blanchard, Jr., not with GM, inventors in patent 2,893,495 for an aircraft power system; and in patent 2,893,496 for a propeller cuff assembly.

• **Charles J. McDowall*,** and **Victor W. Peterson*,** inventors in patent 2,893,525 for an aircraft power system.

• **John W. Rhodes,** (*B.S.Chem.E., Purdue University, 1921*) senior designer, inventor in patent 2,896,474 for a motion transmitting connection.

• **William J. Purchas, Jr.,** (*B.S.M.E., The Detroit Institute of Technology, 1933*) chief engineer, Bearings Department, Transmission Operations; Paul F. Chenea, not with GM; and **Robert D. Bremer,** (*B.S.E.E., Purdue University, 1934*) senior project engineer, Frigidaire Division, inventors in patent 2,897,467 for a sheathed tubular electrical heater.

• **Virgil K. Elder,** senior project engineer; **Thomas A. Heath,** welder; **James F. McLaughlin,** administrator, Experimental Shop Staff Operations; **William P. Zimmerman,** (*B.S.M.E., Purdue University, 1940*) section chief on design projects, Power Turbine Engineering Department; and Charles C. Anderson, no longer with GM, inventors in patent 2,898,442 for a manufacture of compressor vane assembly.

• **Robert M. Tuck,** (*B.M.E., General Motors Institute, 1947*) chief development engineer, Transmission Engineering Department, inventor in patent 2,899,846 for an automatic transmission.

***Inventors' names marked with an asterisk have biographical listings noted previously in this issue's Notes About Inventions and Inventors.**

Aeroproducts Operations
Allison Division
Vandalia, Ohio

• **Morris J. Duer,** (*The Ohio State University*) senior designer, inventor in patent 2,894,501 for a control mechanism for pressure operated actuating device.

• **Howard M. Geyer,** (*B.S.I.E., University of Alabama, 1940*) chief research engineer, inventor in patent 2,897,786 for a twin actuator assembly, and in patent 2,898,888 for a pneumatic actuator assembly.

• **Mack O. Blackburn,** (*The Ohio State University*) chief test engineer, inventor in patent 2,900,765 for a shot peening apparatus.

Buick Motor Division
Flint, Michigan

• **Oliver K. Kelley,** (*B.S., Chicago Technical College, 1925, and Massachusetts Institute of Technology*) now technical assistant to the vice president in charge of Defense Systems Division, inventor in patent 2,893,266 for a transmission; patent 2,898,740 for a transmission; in patent 2,903,083 for a vehicle having auxiliary transmission for the operation of accessories; and in patent 2,906,518 for a viscosimeter type governor.

• **Henry W. Boylan,** (*Wayne State University, 1929*) staff engineer; **Harry C. Doane,** now assistant to the vice president in charge of GM Engineering Staff; and **Arman E. McManama,** (*B.S.M.E., Purdue University, 1950*) senior contact engineer, inventors in patent 2,893,700 for heating ventilating and air conditioning systems for vehicles.

• **Archie D. McDuffie,** (*General Motors Institute, 1934*) staff engineer, inventor in patent 2,893,711 for a charge forming means, and in patent 2,894,459 for a fuel pump.

• **Charles R. Hagler,** (*B.S.M.E., University of Michigan, 1936*) administrative assistant to director of research development, inventor in patent 2,893,751 for a vehicle anti-roll device and power steering control, therefor.

• **George R. Bayley,** (*General Motors Institute, 1929*) staff engineer, inventor in patent 2,893,779 for a filler access mechanism.

• **Henry W. Boylan*,** inventor in patent 2,894,441 for a universal air deflector outlet device.

• **Henry W. Boylan*,** and **James E. Mall,** senior layout man, inventors in patent 2,894,444 for air outlet devices.

• **Cecil A. McKinney,** (*General Motors Institute, 1931*) senior contact engineer, inventor in patent 2,898,896 for a heat exchanger means.

• **Leonard M. Morrish,** (*Michigan State University*) staff engineer, and **Lloyd E. Muller,** (*B.S.M.E., University of Kansas, 1929*) director, Experimental Engineering, inventors in patent 2,899,007 for a muffler.

• **Rudolph J. Gorsky,** (*General Motors Institute, 1935*) staff engineer, inventor in patent 2,905,016 for a transmission control.

• **Frank R. L. Daley, Jr.,** (*B.S. in physical and biological sciences, University of Massachusetts, 1940*) staff engineer, inventor in patent 2,906,360 for a vehicle drive means.

• **Charles D. Holton,** (*B.S.M.E. University of Michigan, 1931*) engineer, inventor in patent 2,906,561 for a brake system.

Cadillac Motor Car Division
Detroit, Michigan

• **Carlton A. Rasmussen,** (*B.S.M.E., Purdue University, 1950*) assistant chief engineer, inventor in patent 2,894,443 for a multiple passage supply duct.

• **Charles P. Bolles, Jr.,** (*B.S.M.E., Purdue University, 1956*) project engineer, inventor in patent 2,894,497 for a manifold with variable length ram pipes.

• **Robert T. Doughty,** (*B.M.E., Tri-State College, 1943*) senior project engineer, inventor in patent 2,898,063 for a resilient support.

• **Bruce M. Edsall,** (*B.S.M.E., Wayne State University, 1942*) staff engineer, and Victor C. Moore, no longer with GM, inventors in patent 2,903,912 for an automatic transmission.

• **Daniel M. Adams,** (*Detroit Institute of Technology and University of Michigan*) staff engineer, and William J. Tell retired, inventors in patent 2,905,286 for a grille assembly.

• **Henry S. Kawecki,** (*University of Michigan and Detroit Institute of Technology*) assistant staff engineer, inventor in patent 2,905,287 for a molding clip and screw.

Central Foundry Division
Saginaw, Michigan

• **William S. Hackett,** (*The Ohio State University*) research engineer, inventor in patent 2,899,726 for an apparatus for assembling shell molds.

• **James P. Child,** senior designer, Fabricast Plant; **John Servi,** designer, Fabricast Plant; **George A. Zink,** (*B.S.M.E., Purdue University, 1929*) manager, nonferrous metals operations, Fabricast Plant; and **Rudolph J. Gorsky,** staff engineer, Buick Motor Division, inventors in patent 2,899,725 for a core making apparatus.

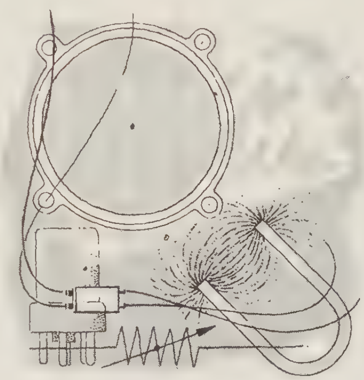
Chevrolet Motor Division
Detroit, Michigan

• **Vernon F. Fishtahler,** (*General Motors Institute, 1940*) design engineer, inventor in patent 2,895,315 for a vibration damping means.

• **Frank C. Burrell,** (*B.S.M.E., University of Wisconsin, 1938*) research engineer, inventor in patent 2,895,447 for liquid level indicators.

• **Frank J. Winchell,** (*Purdue University*) staff engineer, inventor in patent 2,895,-578 for a spring clutch; patent 2,896,478 for a transmission control system; and patent 2,902,126 for a one-way clutch.

• **Russell R. Hershberger,** (*B.S.E.E., Purdue University, 1936*) design engineer; **Shirrell C. Richey,** (*General Motors Institute, 1937*) now with Buick Motor Division; and **Russell J. Bush,** (*B.S.Ch.E., Purdue University, 1925*) project engineer, Inland Manufacturing Division, inventors in patent 2,896,889 for a flexible retainer, and patent 2,896,974 for a bell and spigot joint with plural flexible lip type side.



• **Paul E. Hitch,** (*General Motors Institute, 1939*) staff engineer, and **John W. Walsh,** now with Harrison Radiator Division, inventors in patent 2,899,026 for a cooling system for fluid operated brake.

• **Robert Schilling,** (*M.E. degree, Technical University, Munich, Germany, 1922*) director, research and development, inventor in patent 2,902,104 for a hydraulic power steering system with associated lateral acceleration control means.

• **William S. Wolfram,** (*B.S.M.E., University of Michigan, 1933*) assistant staff engineer, inventor in patent 2,903,036 for a wheel assembly.

• **Philip C. Bowser,** (*B.M.E., The Ohio State University*) now director, research and development, Buick Motor Division, and **Richard E. Denzer,** (*B.M.E. and M.S., The Ohio State University, 1951*) design engineer, inventors in patent 2,906,526 for an air suspension leveling device.

Delco Appliance Division
Rochester, New York

• **Michael A. Dudash,** designer, inventor in patent 2,894,774 for an adjustable length linkage.

• **Kenneth A. Kosbab,** (*B.S.M.E., University of Rochester, 1951*) project engineer, inventor in patent 2,899,901 for a windshield washer pump.

• **Eugene R. Ziegler,** (*B.E.E., University of Rochester, 1943*) design engineer, inventor in patent 2,905,962 for a windshield cleaning system.

Delco Products Division
Dayton, Ohio

• **George W. Jackson,** (*B.S.M.E., Purdue University, 1937*) assistant chief engineer, automotive products, inventor in patents 2,893,504 for a hydraulic steering system with control means for valve reaction pressures; 2,895,743 for a control apparatus for fluid spring suspension; and 2,895,744 for an air-oil suspension unit with ride height control.

• **Joseph M. Rodgers,** (*B.S.E.E., University of Cincinnati, 1938*) laboratory supervisor, and **Robert W. Leland,** (*B.S. E.E., University of Cincinnati, 1937*) staff engineer, inventors in patent 2,895,026 for a switch operating means.

• **William J. Newill,** (*B.S.E.E., Purdue University, 1943*) senior project engineer, inventor in patent 2,898,036 for a windshield washer nozzle assembly.

• **George W. Jackson*,** and **John F. Pribonic,** (*B.S.M.E., Princeton University, 1947*) section engineer, inventors in patent 2,904,330 for a control for fluid suspension system.

• **Frank E. LaFlame,** (*Michigan State University*) senior project engineer, inventor in patent 2,905,372 for a motor compressor unit.

Delco Radio Division
Kokomo, Indiana

• **James H. Guyton,** (*B.S.E.E., 1934, and M.S.E.E., 1935, Washington University*) chief engineer—radio, inventor in patents 2,895,095 for an electronic d-c motor, and 2,903,636 for a transistor inverter and rectifier circuit.

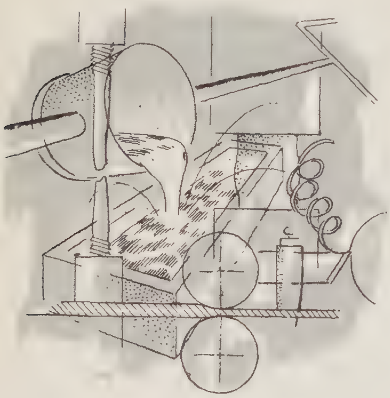
• **Richard L. Jenkins**, (*B.S.E.E., Purdue University, 1944*) senior engineer, inventor in patent 2,896,146 for an oscillator starting circuit.

• **William R. Kearney**, (*B.S.M.E., Purdue University, 1933*) senior project engineer, inventor in patent 2,896,188 for a terminal board.

• **George M. Gaskill**, (*B.S. in physics, Franklin and Marshall College, 1951*) senior project engineer, and **Manfred G. Wright**, (*B.S.M.E., Purdue University, 1938*) head, Mechanical Engineering Section, inventors in patent 2,898,464 for a combined signal seeking push button and manual tuner with dual purpose solenoid.

Delco-Remy Division
Anderson, Indiana

• **Brooks H. Short**, (*B.S.E.E., 1931 and M.S.E.E., 1934, Purdue University*) director, Advanced Engineering, inventor in patent 2,895,090 for a control device.



• **John E. Buskirk**, (*B.S.Ch.E., The Ohio State University, 1948*) section engineer, inventor in patents 2,896,007 and 2,883,-444 for a vent cap and a non-overfill device, respectively.

• **Paul L. Schneider**, (*B.M.E., The Ohio State University, 1921*) special assignment; **Harold J. Cromwell**, (*B.S.M.E., Purdue University, 1933*) senior research engineer; and **William H. Taylor**, special assignment, inventors in patents 2,900,057 and 2,904,148. Both patents for a clutch for engine starting device.

• **Harold J. Cromwell***, inventor in patent 2,900,058 for a clutch for engine starting device.

• **Perry W. House**, (*B.S.Ch.E., Purdue University, 1933*) manufacturing manager, Anderson plants; **George B. Shaw**, supervisor, engineering laboratories; and **Bruce D. Gribben**, (*B.S.Met.E., Purdue University, 1949*) sales engineer, inventors in patent 2,902,125 for an engine starting apparatus.

• **Robert B. Harruff**, (*B.S.M.E., Purdue University, 1940*) research engineer, inventor in patent 2,904,763 for an induction coil.

• **Lyman A. Rice**, (*B.S.E.E., University of Utah, 1935 and M.S.E., University of Michigan, 1936*) staff engineer, inventor in patent 2,906,939 for a regulating circuit for generator.

• **Don Key**, chief machine designer, Process Engineering Department, and **Donald G. Mahoney**, (*M.E., Purdue University, 1933*) process engineer, inventors in patent 2,880,896 for a storage battery and method of making same.

• **Donald P. Gentry**, (*B.S.Ch.E., Purdue University, 1944*) senior experimental engineer, Delco Battery plant, Muncie, and **Charles R. Shepherd**, (*B.S.Ch.E., The Ohio State University, 1948*) senior reliability engineer, inventors in patent 2,883,358 for hard rubber compound containing a mixture of portland cement and petroleum coke.

• **Charles E. Buck**, (*B.S.E.E., Purdue University, 1935*) engineer—solenoids, inventor in patent 2,886,720 for a coordinator for windshield washer and wipers.

• **Robert P. MacKenzie**, (*A.B.Chem., Indiana University, 1948, and B.S.Met.E., Purdue University, 1950*) experimental metallurgist, inventor in patent 2,887,522 for a storage battery and method of making same.

• **Charles E. Bates**, (*B.S.Met.E., University of Illinois, 1943*) engineer, inventor in patent 2,889,392 for a vent cap.

• **Paul C. Kline, Jr.**, (*General Motors Institute, 1949*) section head, ignition equipment, inventor in patent 2,889,418 for an ignition distributor.

• **Garth A. Rowls**, (*B.S.M.E., Purdue University, 1932*) product engineer—storage batteries, and **Thomas L. Kendall**, (*B.S.Ch.E., Purdue University, 1933*) United Motors Service, inventors in patent 2,890,262 for a storage battery.

• **Robert A. Cheetham**, (*B.S.M.E., Purdue University, 1951*) section head, d-c motors, inventor in patent 2,892,454 for engine starting apparatus.

Detroit Diesel Engine Division
Detroit, Michigan

• **Vernon Schafer, Jr.**, (*B.S.E.E., University of Michigan, 1938 and M.S.M.E., Case Institute of Technology, 1939*) staff engineer, inventor in patent 2,893,371 for an expansion joint.

• **Albin Chaplin**, (*B.S.M.E., Detroit Institute of Technology, 1948*) senior designer, and **Virgin C. Reddy**, (*B.S.M.E., 1934, and M.S.M.E., Iowa State College, and General Motors Institute, 1936*) development engineer, inventors in patent 2,894,498 for a multiple speed drive and control system.

• **Michael H. O'Brien**, (*University of Detroit*) field test engineer, and **Roger D. Wellington**, (*B.S.M.E., University of Rochester, 1930*) director of test, Engineering Laboratory, inventors in patent 2,894,502 for an engine protective device.

Detroit Transmission Division
Ypsilanti, Michigan

• **Forrest R. Cheek**, (*B.S.M.E., University of Illinois, 1946*) senior project engineer, and **Norman W. Reighard**, (*B.M.E., General Motors Institute, 1949*) assistant staff engineer, inventors in patent 2,896,468 for automatic plural step ratio transmissions.

These patent listings are informative only and are not intended to define the coverage which is determined by the claims of each one.

• **Darrel R. Sand,** (*B.M.E., General Motors Institute, 1949*) assistant staff engineer, and **Thomas R. Zimmer, Jr.,** (*B.M.E., General Motors Institute, 1954*) project engineer, inventors in patent 2,898,738 for a hydrodynamic drive device.

• **Darrel R. Sand*,** inventor in patent 2,904,150 for a clutch plate.

Diesel Equipment Division
Grand Rapids, Michigan

• **Donald E. Wortman,** (*B.S.M.E., University of Michigan, 1949*) project engineer, inventor in patent 2,893,647 for an adjustable fuel nozzle.

• **Ronald C. Groves,** (*B.E.E., 1936 and B.M.E., 1937, Lawrence Institute of Technology*) now at Rochester Products Division, inventor in patents 2,895,461 and 2,899,948 for a fuel injection system, and 2,902,989 for a charge forming means.

• **Conrad A. Teichert,** (*M.E., Mittweida Technical College, Germany, 1922, and Aero. E., University of Berlin, 1933*) senior designer, inventor in patent 2,898,051 for a fluid injection device.

• **Elias W. Scheibe,** (*B.S.M.E., University of Michigan, 1940*) senior project engineer, inventor in patent 2,902,015 for a hydraulic lash adjuster.

Electro-Motive Division
La Grange, Illinois

• **Torsten O. Lillquist,** electrical research engineer, inventor in patent 2,898,582 for a wheel lock and wheel slide detection system.

• **Robert L. Dega,** (*B.M.E., General Motors Institute, 1948*) now supervisor, Mechanical Development Department, GM Research Laboratories, inventor in patent 2,898,037 for a centrifuge for clarifying fluid, and 2,900,129 for a centrifuge for cleaning fluids.

• **Arthur H. Juhlin,** senior project engineer, and **Albert L. Macan,** (*International School of Correspondence*) air brake engineer, inventors in patent 2,900,043 for a filter.

GM Engineering Staff
Detroit Michigan

• **Raymond J. Haefner,** (*B.M.E., General Motors Institute, 1950*) now at Rochester Products Division, inventor in patent 2,893,365 for a fuel injection means.

• **Richard P. Barnard,** (*B.S.M.E., University of Michigan, 1947 and Bachelor of law degree, George Washington University, 1950*) patent attorney, Patent Section, Detroit Office, inventor in patent 2,893,-737 for power actuated closure.

• **Ralph S. Johnson,** (*B.S.M.E., Purdue University, 1933*) senior project engineer, inventor in patent 2,894,414 for a connecting rod mounting.

• **Albert J. Zupancic,** senior technician, inventor in patent 2,894,735 for a fuel metering system.

• **Gilbert K. Hause,** engineer in charge, Transmission Development Group, inventor in patent 2,896,471 for a transmission control.

• **Gilbert K. Hause*, Clifford C. Wrigley,** (*B.S.M.E., University of Colorado, 1936 and Yale University*) assistant engineer in charge, Transmission Development Group; and **Oliver K. Kelley,** technical assistant to the vice president in charge of Defense Systems Division, inventors in patent 2,898,644 for a windshield cleaning mechanism.

• **Lewis D. Burch,** (*B.S.M.E., Purdue University, 1923, George Washington University, Akron Law School, and Wayne State University*) patent attorney, Patent Section, Detroit Office, inventor in patent 2,898,898 for an engine.

• **Lloyd M. Keighley,** patent attorney, Patent Section, Dayton Office, inventor in patent 2,903,865 for an ice block releaser and storage bucket.

• **Von D. Polhemus,** (*B.S.M.E., University of Cincinnati, 1933*) engineer in charge, Structure and Suspension Development Group, inventor in patent 2,906,543 for an interconnected torsion bar suspension.

• **Lothrop M. Forbush,** (*B.S., Harvard University, 1939, and Massachusetts Institute of Technology*) engineer in charge, Vehicle Development Group, inventor in patent 2,906,558 for a vehicle wheel fastening means.

• **John S. Wroby,** (*B.M.E., University of Detroit, 1942*) design engineer, inventor in patent 2,906,572 for a drive shaft vibration isolation mount.

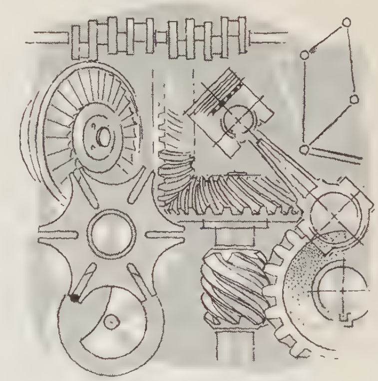
Euclid Division
Cleveland, Ohio

• **Edward R. Fryer,** (*General Motors Institute, and B.S.M.E., Massachusetts Institute of Technology, 1945*) senior project engineer, inventor in patent 2,893,139 for a double ratio lever mechanism.

• **Henry A. Peller,** chief product engineer, inventor in patents 2,893,470 and 2,894,562 for an adjustable seat and vehicle seats, respectively.

• **Robert H. Scott,** (*B.S.M.E., Royal Technical College, Glasgow, Scotland, 1951*) component engineer, inventor in patent 2,894,416 for a fluid lock differential.

• **Raymond Q. Armington,** (*B.I.E., The Ohio State University, 1928*) general manager, and **Arthur P. Armington,** (*B.S.M.E., Case Institute of Technology*) staff engineer, inventors in patent 2,899,760 for a multi-purpose bulldozer blade.



• **Arthur P. Armington*,** inventor in patents 2,901,846 for a four-wheel drive tractor grader, and 2,904,910 for an angle bulldozer for four-wheel steer tractor.

• **Ralph J. Bernotas,** (*B.S.M.E. and M.S.M.E., Case Institute of Technology*) senior project engineer, inventor in patent 2,880,746 for a combination accumulator and unloading valve.

• **Paul Pleska,** (*Case Institute of Technology and Fenn College*) designer, inventor in patent 2,891,331 for a transport dolly.

Fisher Body Division
Detroit, Michigan

• **Lewis J. Lamm,** (*General Motors Institute, 1937; B.S.M.E., 1938, and Juris Doctor of Law degree, 1941, George Washington University; U. S. Naval Academy postgraduate school; and M.S. in communication engineering, Harvard University, 1946*) assistant plant manager, Grand Rapids plant No. 2, inventor in patent 2,894,175 for an apparatus for spray painting.

• **Robert M. Fox,** (*B.M.E., General Motors Institute, 1950*) senior project engineer, inventor in patent 2,894,302 for a spring clip.

• **James H. Wernig,** (*Johnson School of Body Design and Engineering, and Alexander Hamilton Institute*) general director of engineering and related activities, inventor in patent 2,895,763 for a vehicle body structure.

• **Louis P. Garvey,** (*B.M.E., University of Detroit, 1940*) assistant engineer in charge, Product Engineering Activity, and **Clyde H. Schamel,** (*B.S.E.E., University of Notre Dame, 1927*) engineer in charge, Central Experimental and Development Department, inventors in patent 2,896,990 for a vehicle closure latch.

• **Peter M. Hart,** (*University of Mississippi*) senior project engineer, and **Aloys C. Hollerbach,** engineer in charge, Body Engineering Department, inventors in patent 2,897,000 for a pivoted arm rest assembly for vehicle body.

• **Joseph L. Lelli,** (*B.M.E., General Motors Institute, 1948*) assistant engineer in charge, Experimental and Development Department, and **Samuel C. Pollock,** (*University of Detroit*), senior design engineer, GM Styling Staff, inventors in patent 2,897,003 for a convertible top linkage and actuating means.

• **Douglas T. Bull,** (*Westham Technical College and Southeast Essex Technical College, London, England*) senior project engineer, and **Arkady A. Lobanoff,** (*Wayne State University and University of Michigan*) engineer in charge, front end and instrument panels, inventors in patent 2,897,533 for grommets, bushings and the like.

• **Neal E. O'Connor,** (*B.S.M.E., University of Michigan, 1952*) senior project engineer, inventor in patent 2,898,712 for a work tool.

• **Louis P. Garvey*; James D. Leslie,** (*B.M.E., University of Detroit, 1939*) engineer in charge, Mechanical Department; and **Wilson H. West,** (*University of Detroit*) senior project engineer, inventors in patent 2,900,183 for an operating means for a window regulator mechanism.

• **Roy T. Collins,** (*General Motors Institute, 1939*) senior project engineer; **Arthur R. Dey,** senior staff assistant; and **Joseph L. Winston,** (*B.A., University of Michigan, 1934*) engineer in charge, material handling development and plant contact, inventors in patent 2,903,217 for a material handling means.

• **Carl E. Hedeen,** (*Wayne State University*) senior engineer in charge, Body Engineering Activity—Design and Drafting, inventor in patent 2,902,277 for window regulator.

• **Joseph G. Joachim,** senior drafting room checker, and **Clayton H. White,** (*B.M.E., Lawrence Institute of Technology, 1953*) project engineer, inventors in patent 2,903,288 for a latch striker mechanism.

• **Engelbert A. Meyer,** senior project engineer, inventor in patents 2,903,939 for a stud retainer with teeth arranged to engage stud in a plurality of positions and 2,906,159 for a threadless stud retainer having a plurality of lugs to engage work surface.

• **Stanley D. Cockburn,** (*Lawrence Institute of Technology*) senior designer, and **Robert M. Fox*,** inventors in patent 2,904,365 for a vehicle door latch.

• **William A. Brady, Jr.,** (*Lawrence Institute of Technology*) senior process engineer, inventor in patent 2,905,803 for a nut loading and welding machine.

Frigidaire Division
Dayton, Ohio

• **John Weibel, Jr.,** (*B.S.M.E., Louisiana State University, 1948, and M.S.M.E., Purdue University, 1950*) senior project engineer, inventor in patent 2,893,626 for a refrigerating apparatus.

• **Verlos G. Sharpe,** (*B.S.M.E., Purdue University, 1948*) section engineer, inventor in patent 2,893,726 for a refrigerator door control.

• **George B. Long,** (*B.S.E.E., Purdue University, 1937*) supervisor of major product line, and Ray E. Clever, not with GM, inventors in patent 2,894,105 for a domestic appliance.

• **Orson V. Saunders,** supervisor, major products, and **Carl F. Petkwitz,** (*B.S.M.E., University of Dayton, 1925*) engineer, inventors in patent 2,894,378 for a refrigerating apparatus.

• **Orson V. Saunders*,** inventor in patent 2,894,379 for a refrigerating apparatus.

• **Byron L. Brucken,** (*B.S. degree, University of Dayton, 1956*) senior project engineer, inventor in patent 2,894,698 for a vertical support shaft for motor rotor armature and comminutor impeller disc.

• **Clifford H. Wurtz,** (*B.S., University of Illinois, 1929*) manager, Refrigerated Appliances Engineering, inventor in patent 2,894,731 for a refrigerating apparatus.

• **George B. Long*,** inventor in patent 2,895,320 for a washer.

• **Carl M. Schell,** (*Wittenberg College, and University of Chicago*) general supervisor of drafting, Research and Future Products Department, inventor in patent 2,896,312 for a refrigerating apparatus.

• **Lester M. Miller,** (*Dayton Art Institute, Art Academy of Cincinnati, and The Central Academy of Commercial Art*) junior engineer, and **Robert F. Bentley,** GM Process Development Staff, inventors in patents 2,897,665 and 2,897,670 for a washing machine and a single indicator air gage circuit, respectively.

• **James W. Jacobs,** (*B.S.M.E., University of Dayton, 1954*) manager, research and future products engineering, inventor in patent 2,899,253 for a domestic appliance.

• **Jesse L. Evans,** (*B.S.M.E., University of Dayton, 1943*) senior project engineer, inventor in patent 2,899,255 for a range with pull-out shelf.

• **Curtis P. Kelley,** (*B.S.M.E., University of Kentucky, 1936*) supervisor, dishwasher and food waste disposer section, Non-Refrigerated Appliances Engineering Department, inventor in patent 2,899,256 for a refrigerating apparatus.

• **Clifford H. Wurtz***, and **Leonard J. Mann,** (*M.E., University of Cincinnati, 1940*) senior project engineer, inventors in patent 2,900,806 for a self-defrosting two-temperature refrigerator.

• **Keith K. Kesling,** (*University of Dayton and Dayton Art Institute*) project and design engineer, inventor in patent 2,902,952 and 2,903,711 for a refrigerator cabinet safety door and a domestic appliance, respectively.

• **Ronal H. Whyte,** (*Sinclair College, 1935*) senior engineer, and **Richard T. Clement,** GM Styling Staff, inventors in patent 2,904,897 for a hamper bag for laundry appliances.

• **Jacob Smilack,** (*B.S.M.E., Purdue University, 1925*) senior project engineer, inventor in patent 2,904,972 for a room conditioner.

• **George B. Long***, and **Byron L. Brucken***, inventors in patent 2,906,111 for a washing machine.

• **Floyd O. Moody,** (*B.S., Otterbein College, 1944 and B.E.E., The Ohio State University, 1949*) project engineer, and David H. Siebenthaler, not with GM, inventors in patent 2,906,132 for a toggle.

GMC Truck and Coach Division
Pontiac, Michigan

• **Richard H. Hancock,** (*University of Michigan*) manual transmission engineer, inventor in patent 2,893,104 for an air suspension.

• **Adam S. Lamont,** general foreman, Experimental Machine Shop, inventor in patent 2,895,575 for a variable speed mechanism.

• **Donald J. La Belle,** (*B.M.E., Univerversity of Detroit, 1939*) staff engineer, inventor in patent 2,902,240 for a mounting means.

Guide Lamp Division
Anderson, Indiana

• **Harold E. Todd,** (*B.S.E.E., Purdue University, 1940*) senior project engineer, inventor in patent 2,894,172 for a reversing switching circuit for automatic headlight.

• **Howard I. Slone,** (*B.S.E.E., Purdue University, 1938*) accessory engineer, inventor in patent 2,902,902 for a remote control mirror assembly.

• **Eugene G. Matkins,** (*B.M.E., General Motors Institute, 1952*) senior project engineer—reliability, and Carl E. Atkins, no longer with GM, inventors in patent 2,904,699 for an automatic headlight dimmer system.

• **Eugene G. Matkins***, and **Charles W. Miller,** (*Purdue University*) senior project engineer, inventors in patent 2,905,867 for a pulsed light sensitive control circuit and 2,905,868 for a self oscillating light controlled circuit.

Inland Manufacturing Division
Dayton, Ohio

• **Arthur J. Frei,** senior project engineer and **Raymond C. Davis,** (*M.E., Dayton Night College, 1920*) administrative engineer, inventors in patent 2,895,312 for an ice cube tray.

• **George Rappaport,** (*B.S. in chemistry, Miami University, 1944, and M.S. in chemistry, The Ohio State University, 1950*) senior chemist; **Joseph A. Szaruga,** (*B.S. in Agr. Biochemistry, The Ohio State University, 1953*), development chemist; and **James R. Wall,** (*B.Ch.E., University of Dayton, 1937 and M.Ch.E., Cornell University, 1939*) supervisor, Advanced Development Laboratory, inventors in patent 2,895,926 for preparation of polyurethane foam and reaction mixture therefor.

• **Russell J. Bush,** (*B.S.Ch.E., Purdue University, 1925*) project engineer, and **Robert E. Sayre,** (*Capitol University and The Ohio State University*) sales engineer, inventors in patent 2,898,643 and 2,898,647 for a window frame.

• **Harold J. Reindl,** (*B.Ch.E., University of Dayton, 1942*) section head, Paints, Coatings, and Adhesives Laboratory, inventor in patent 2,899,136 for an electrostatic painting apparatus.

• **George W. Beck,** (*B.M.E., General Motors Institute, 1947*) assistant chief engineer, product design; **Walter Ziffer,** (*B.S. M.E., Vanderbilt University, 1954*) senior project engineer; and **William E. Sehn,** (*B.M.E., General Motors Institute, 1951*) engineer in charge, Design and Drafting Department, Fisher Body Division, inventors in patent 2,902,732 for a weather-strip retaining clip.

• **Allen L. Everitt,** (*M.S.M.E., Purdue University, 1931*) section engineer, inventor in patent 2,903,208 for a resilient mounting.

• **James R. Wall***, and **Robert D. Powell,** (*B.I.E., General Motors Institute, 1952*) development engineer, inventors in patent 2,904,063 for a flow control mechanism.

Moraine Products Division
Dayton, Ohio

• **William F. Erickson,** (*General Motors Institute, 1943*) section engineer, inventor in patent 2,894,525 for a combined flow control and relief valve.

New Departure Division
Bristol, Connecticut

• **William P. Burroughs,** (*Northeastern Polytechnic Institute*) experimental research chemist, inventor in patent 2,893,181 for hydrogen elimination in treatment of metals.

GM Overseas Operations Division
New York, New York

• **Leslie G. Burgess,** senior project engineer, and **William H. Gee,** (*Coventry Technical*) manager, sheet metal production engineering, Vauxhall Motors, Limited, Luton, England, inventors in patent 2,902,109 for silencers for pulsating gaseous currents.

• **Kenneth E. Buckman,** assistant chief engineer, No. 2 Plant, AC-Delco Division, GM Limited, London, England, inventor in patent 2,903,136 for fluid filters.

Pontiac Motor Division
Pontiac, Michigan

• **George T. Timoff,** (*B.S.E.E. and B.S. in Mathematics, University of Michigan, 1953*) senior project engineer, and **William K. Jensen,** (*Wayne State University and Lawrence Institute of Technology*) electrical development engineer, inventors in patent 2,894,090 for a horn blowing mechanism.

• **Francis H. Grady,** (*International Correspondence School*) transmission engineer, inventor in patent 2,896,470 for vehicle controls.

• **John Z. DeLorean,** (*B.S.I.E., Lawrence Institute of Technology, 1948; M.S.A.E., Chrysler Institute, 1952; M.B.A., University of Michigan, 1957; and Detroit College of Law*) assistant chief engineer in charge of advanced design, inventor in patent 2,898,750 and 2,906,105 for a universal joint.

• **Mark H. Frank,** (*B.S.M.E., Michigan State University, 1927*) motor engineer and **Clayton B. Leach,** (*A.B. in mathematics and chemistry, Park College, 1934, and General Motors Institute*) chassis engineer, inventors in patent 2,899,015 for engine bearings and lubrication system.

• **Clayton B. Leach*,** inventor in patent 2,902,014 for a valve actuating mechanism for engines and 2,902,021 for a cylinder block.

• **Richard M. Gorman,** group leader, and **Clayton B. Leach*,** inventors in patent 2,902,102 for a bumper exhaust.

• **George W. Lampman,** (*International Correspondence School*) project engineer, and **L. Ray Sampson,** (*General Motors Institute, 1932*) electrical engineer, inventors in patent 2,904,769 for a spark plug nipple.

GM Research Laboratories
Detroit, Michigan

• **William C. Bubniak** (*B.S.M.E., Lawrence Institute of Technology, 1949*) senior design engineer, inventor in patent 2,893,699 for a regenerator and seal therefor.

• **Joseph T. Wentworth,** (*B.S.M.E., University of Michigan, 1952*) senior research engineer, inventor in patents 2,894,734 for a carburetor venting device; 2,894,736 for a carburetor vent system, and 2,895,292 for a fuel and exhaust gas combustion control of an internal combustion engine.

• **Robert F. Thomson,** (*B.S.M.E., 1937; M.S.M.E., 1940; and Ph.D., 1941, University of Michigan*) head, Metallurgical Engineering Department, inventor in patent 2,894,319 for a sintered powdered aluminum base bearing.

• **Joseph L. Greene,** (*B.A. in chemistry, Wayne State University, 1956*) research chemist, and **James C. Holzwarth,** (*B.S.M.E., 1945, and M.S.M.E., 1948, Purdue University*) supervisor, Metallurgical Engineering Department, inventors in patent 2,894,850 for a method of galvanizing ferrous metal strip.

• **Claude J. Kinsey,** general supervisor, Drafting and Design, Special Problems Department, and **John H. Varterasian,** (*B.S.E.E., University of Michigan, 1952*) research engineer, inventors in patent 2,898,764 for a balancing machine.

• **James C. Holzwarth*,** and **Robert F. Thomson*,** inventors in patent 2,899,304 for a highly wear resistant zinc base alloy.

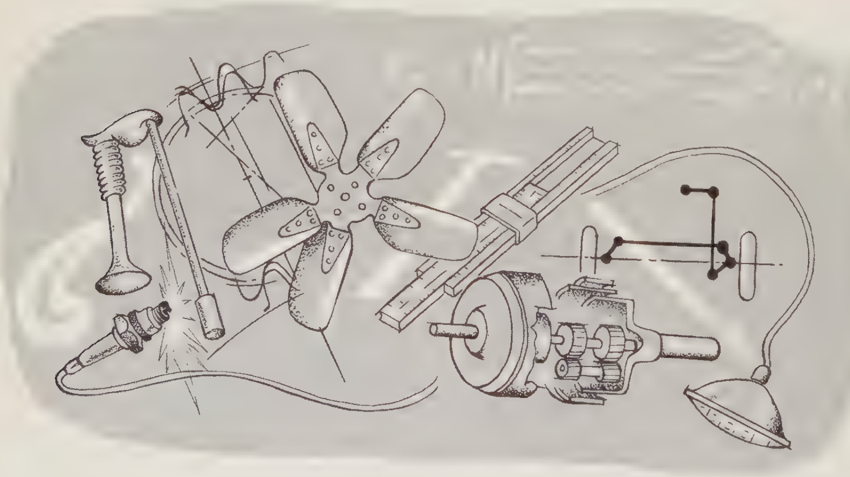
• **Alexander Somerville,** (*Ph.D., Northwestern University, 1950*) supervisor, Isotope Laboratory, inventor in patent 2,903,590 for a nuclear radiation measuring instrument.

• **James C. Holzwarth*,** and **Dean K. Hanink*,** now chief metallurgist, Allison Division, inventors in patent 2,903,785 for a method of hot working titanium.

• **Joseph B. Bidwell,** (*B.S.M.E., Brown University, 1942*) head, Engineering Mechanics Department, inventor in patent 2,904,120 for power steering.

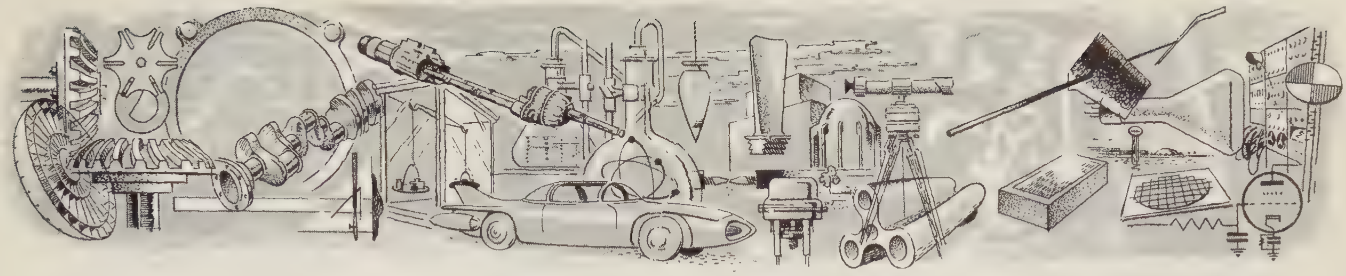
• **Walter E. Sargeant,** (*B.S.E.E., University of Michigan, 1926*) senior research engineer, inventor in patents 2,905,832 and 2,905,904 for a power supply means and timing control circuit, respectively.

• **Charles W. Vigor,** (*B.S.Met.E., University of Michigan, 1950*) senior metallurgical engineer, and Alfred L. Boegehold, retired, inventors in patent 2,906,-008 for brazing of titanium members.



Rochester Products Division
Rochester, New York

• **Lawrence C. Dermond,** (*Purdue University and Tri-State College*) staff engineer, inventor in patents 2,894,499 for a fuel control system; 2,899,950 for a fuel injection acceleration enrichment device; and 2,900,167 for a fuel injection metering valve.



• **Lawrence C. Dermond***, and **Elmer Olson**, (*Lewis Institute*) engineering consultant, inventors in patent 2,898,096 for a fuel injection system.

• **Elmer Olson***, inventor in patent 2,899,194 for carburetor fuel nozzle construction.

• **Lawrence C. Dermond***, **Ellsworth A. Kehoe**, (*B.S.E.E., University of Alabama*) assistant chief engineer, Product Engineering Department; and **Elmer Olson***, inventors in patent 2,899,941 for a fuel cut-off device for fuel injection system.

• **Clarence R. Lunn**, supervisor, Michigan area sales engineers, inventor in patent 2,902,045 for a carburetor float mechanism.

Saginaw Steering Gear Division
Saginaw, Michigan

• **Henry S. Smith**, (*B.S., Central Michigan University, 1938, and M.A., University of Michigan, 1940*) administrative engineer; **Philip B. Zeigler**, (*B.S.M.E., Purdue University, 1941*) chief engineer; and C. W. Lincoln, retired, inventors in patent 2,897,684 for an in-line hydraulic power steering gear.

• **Joseph J. Verbrugge**, (*General Motors Institute, 1941*) now development engineer, Buick Motor Division, inventor in patent 2,902,829 for a hydraulic power brake unit.

• **William Blair Thompson** (*B.M.E., General Motors Institute, 1950*) on leave of absence, and Ray J. Wrobbel, not with GM, inventors in patent 2,905,459 for high pressure fluid seals.

GM Styling Staff
Detroit, Michigan

• **Robert McClure**, (*Rolls Royce, Ltd., 5-year engineering program*) senior project engineer, inventor in patent 2,895,345 for a safety steering wheel.

• **John Himka**, (*diploma in Aero.E., Academy of Aeronautics, 1941*) general supervisor, Body Development Studio, and Frederick C. Walther, retired, inventors in patent 2,895,764 for a vehicle folding top.

• **Gelasio F. Garcia**, (*Havana University, Cuba*) senior designer, Chevrolet Studio, and **Joseph R. Schemansky**, chief designer, Preliminary Design Studio, inventors in patent 2,896,997 for an automotive vehicle body.

• **Delbert C. Probst**, (*Milliken University, 1922*) senior design engineer, inventor in patent 2,897,916 for a cleaning and silencing means.

• **Harry A. Mackie**, (*B.S.M.E., and B.S.Aero.E., Louisiana State University*) design engineer, inventor in patent 2,903,904 for an adjustable steering column and shaft.

• **Samuel C. Pollock**, (*University of Detroit*) senior design engineer, Exterior Engineering Department, inventor in patent 2,893,782 for attachment of convertible top fabric to side rail.

Ternstedt Division
Detroit, Michigan

• **Barthold F. Meyer**, (*B.S.M.E., Pratt Institute, 1939, and Johns Hopkins University*) engineering group supervisor, inventor in patents 2,893,258 and 2,899,832 for a mechanical movement device and a window regulator drive mechanism, respectively.

• **William G. Hoag**, (*B.S.C.E., University of Michigan, 1929*) design group leader, and **Robert C. Liem**, (*B.S. in industrial management, University of Kansas, 1950*) design group leader, inventors in patent 2,893,781 for a window guide mechanism.

• **Edmund S. Badura**, (*B.M.E., University of Detroit, 1951*) design group leader, inventor in patent 2,895,161 for a counter-balancing hinge assembly.

• **Barthold F. Meyer***, and **LaVerne B. Ragsdale**, (*University of Detroit, Franklin College, and B.S.M.E., Lawrence Institute of Technology, 1939*) staff engineer, inventors in patent 2,905,003 for a window regulator drive mechanism.

• **Thomas E. Lohr**, (*Tri-State College*) senior design engineer, and **George W. Sierant**, (*B.S.M.E., Lawrence Institute of Technology, 1947*) design group leader, inventors in patent 2,905,012 for a mechanical movement device.

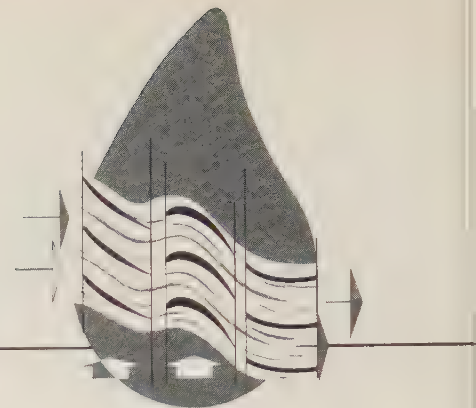
• **John P. Bogater**, (*B.S.M.E., Detroit Institute of Technology, 1937*) group design leader, and **LaVerne B. Ragsdale***, inventors in patent 2,894,277 for a deck lid hinge with adjustable counterbalance.

• **Albert J. Colautti**, (*B.M.E., Detroit Institute of Technology, 1952*) design group leader, inventor in patent 2,888,293 for a vehicle door latch.

• **Frank A. Croskey**, research engineer, and **Charles D. Tuttle**, (*Ph.D., Michigan State University, 1933*) senior experimental chemist, inventors in patent 2,890,388 for an electrostatic spray charger.

A Typical Problem in Engineering:

Determine the Angular Setting of Spokes Used in the Second Turbine of an Automatic Transmission



The second turbine used in one type of automatic transmission is mounted on a supporting hub structure containing three streamline shaped spokes. These spokes rotate through the flow path of the torque converter and transmit a driving torque to an output shaft. To present the least resistance to fluid flow, the spokes must be given the proper angular orientation relative to the torque converter blading. The problem presented here is to determine the angular orientation the spokes should have to give minimum drag loss under a set of specified operating conditions.

Good engineering design requires the resolution of technical, economic, space, and weight factors into the best composite package capable of performing the required function. The automatic transmission is typical of this type of design. Frequently, it is desirable to affect a design balance in the performance of a particular component in order to achieve a better overall arrangement and function of the various components, which as a composite will provide good performance characteristics and reliability at a competitive price.

An example of such a design balance is the type of supporting hub used for the second turbine in one type of automatic transmission (Fig. 1). This structure

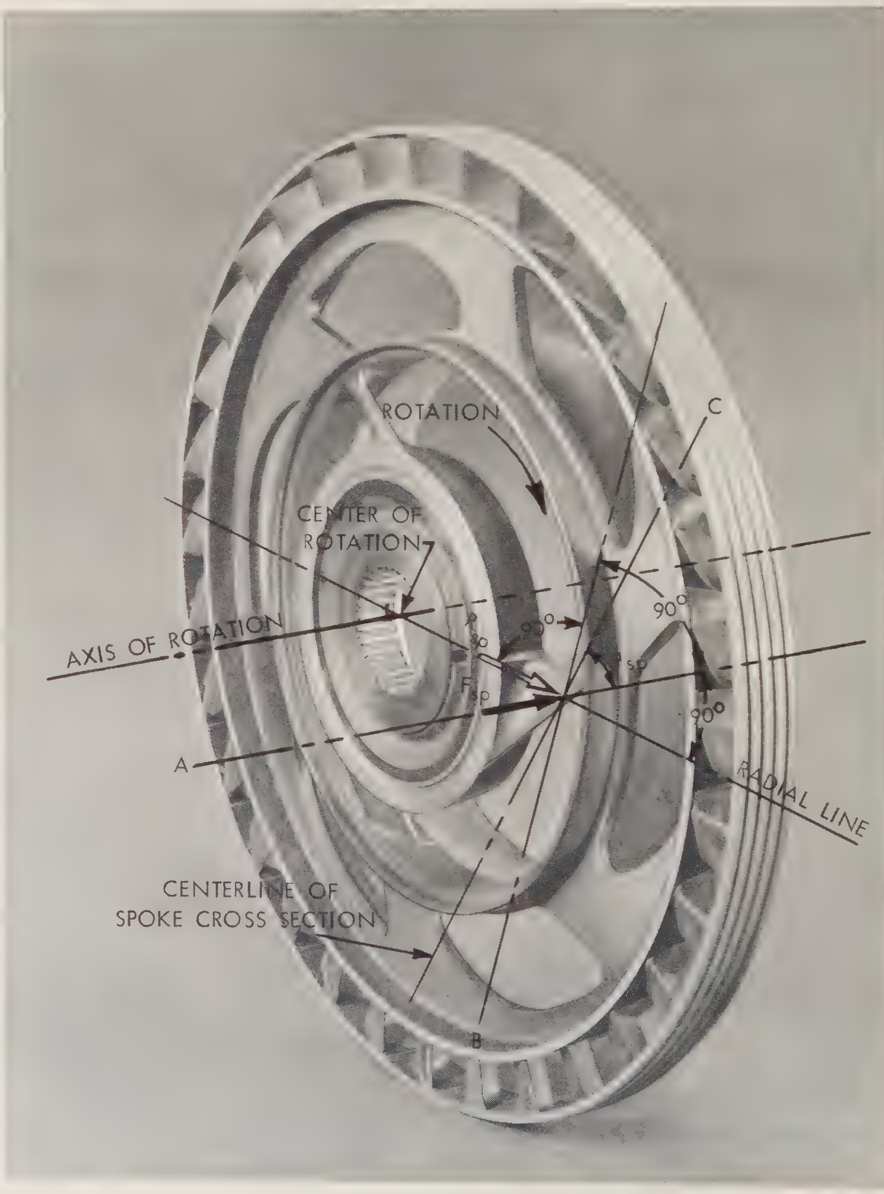


Fig. 1—The second turbine of one type of automatic transmission is an aluminum die casting mounted on three, tangentially arranged, streamline shaped spokes which transmit torque, produced by hydraulic forces on the blade cascade, to an output shaft. These spokes must be properly oriented relative to the converter blade system in order to present the least resistance to hydraulic flow and obtain the best overall performance from the torque converter. The problem presented here deals only with the orientation of the centerline of the cross section relative to the axis of rotation of the turbine. The angle a_{sp} is measured in a plane parallel to the axis and perpendicular to the radial line through the point of intersection of the design path and the leading edge of the spoke. Flow velocity is in the direction indicated by F_{sp}. Lines *A* and *B* represent axes of the plane of the spoke cross section. Line *C* represents the centerline of the spoke cross section. The radius of the spoke is defined as R_{sp}. It can be seen that the long axis of the spoke does not have a pure radial orientation. The long axis is approximately tangential to the hub at the spoke inner radius in order to achieve maximum strength as well as minimum drag loss. The location of the spoke axis can form the basis for another interesting problem.

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General Motors
Engineering Staff

Assisted by Wesley J. Trathen
General Motors Institute

Set the spokes
to minimize
drag loss

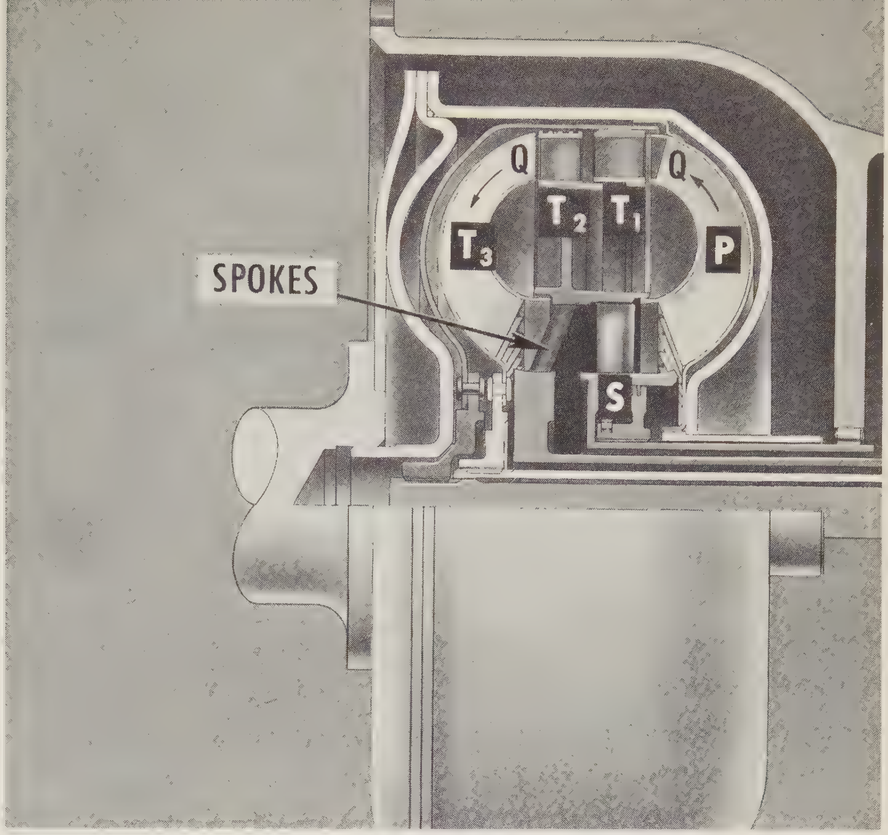


Fig. 2—Shown here is the arrangement of the basic torque and gear components in one type of automatic transmission. The spokes of the supporting hub (Fig. 1), on which is mounted the second turbine T_2, are located in the flow path of the torus between the exit of the third turbine T_3 and the entrance to the stator *S*. The converter pump *P* is driven directly from the engine flywheel. The first turbine T_1 and second turbine rotate at their particular geared speeds relative to the converter output shaft. Each of these turbines is provided with a one-way clutch, or free wheel unit, which allows them to rotate freely in the oil stream when they are not providing driving torque to the output shaft. The third turbine rotates at the same speed as the transmission output shaft when the transmission is in the driving range. The direction of the volume rate of flow, or circulation rate, is indicated by *Q*.

makes use of three spokes to transmit hydraulic driving torque to an output shaft. The spokes of the supporting hub structure cross the flow path of the torque converter between the third turbine discharge and the stator entrance (Fig. 2). The spokes pass through the hydraulic flow path of the torque converter and the hydraulic driving force on the blade cascade at the outer part of the converter torus is transmitted as a torque to the output shaft.

While these spokes transmit driving torque they also offer an obstruction to fluid flow. This obstruction can be minimized by streamline shaped spokes, of sufficient strength, oriented in a specific direction in the hydraulic flow path.

Problem

The problem is to determine the correct angular orientation of the spokes, relative to the axis of rotation of the turbine, which will minimize drag loss at specified conditions of operation. In defining the conditions of operation, it can be assumed that the integrated effect of the fluid acts at one radius point in any portion of the torus at a corresponding angular direction. The theoretical line formed by the radius points in the radial plane through the centerline of rotation is called the *design path.* Experience indicates that this simplifying assumption is satisfactory for establishing the initial design of the spokes.

The specified conditions of operation and the physical dimensions to be used are as follows:

- Q = the volume rate of flow, or the circulation rate, in a radial plane containing the axis of rotation (cu ft per min). The flow areas A_t, A_p, A_s, and A_{sp}, given below, are defined as being perpendicular to Q at a particular point in the torus (Fig. 2).
- Converter output speed = third turbine speed = N_{T3} = 1,508 rpm
- Converter input torque = T_p = 150 lb-ft
- Converter speed ratio = $(N_{T3} \div N_p) = 0.8$
- Pump discharge radius = R_p = 0.445 ft
- Pump discharge area = A_p = 0.142 sq ft
- Pump discharge angle = $a_p = +45°$
- Turbine discharge radius = R_p = 0.227 ft
- Turbine discharge area = A_t = 0.121 sq ft
- Turbine discharge angle = $a_t = -42°$
- Stator discharge radius = R_s = 0.220 ft
- Stator discharge area = A_s = 0.119 sq ft
- Stator discharge angle = $a_s = +40°$
- Design path radius at leading and trailing edges of the spoke = R_{sp} = R_s = 0.220 ft
- Flow path area at the leading and trailing edges of the spokes = A_{sp} = A_s = 0.119 sq ft
- Density of the fluid = d = 48.9 lb per cu ft
- Gravitational constant = g = 32.2 ft per sec²
- Speed of the second turbine = N_{T2} = $1.63 N_{T3}$
- Speed of the stator = N_s = 0 rpm.

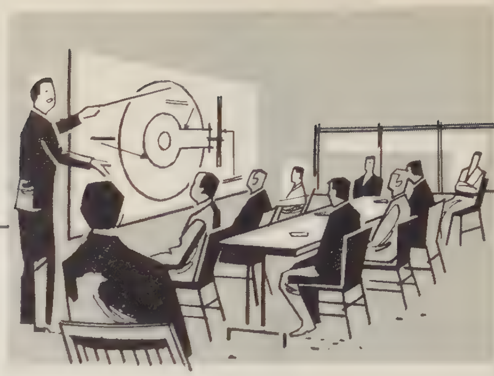
The solution to the problem will be published in the April-May-June 1960 issue of the General Motors Engineering Journal.

Bibliography

Literature useful in the solution of this problem includes the following:

Society of Automotive Engineers Handbook, sections on Hydrodynamic Drive Terminology and Symbols for Hydrodynamic Drives.

Technical Presentations by GM Engineers and Scientists



The technical presentation is another way in which information about current engineering and scientific developments in General Motors can be made available to the public. A listing of speaking appearances by General Motors engineers and scientists, such as that given below, usually includes the presentation of papers before professional societies, lecturing to college engineering classes or student societies, and speaking to civic or governmental organizations. Educators who wish assistance in obtaining the services of GM engineers and scientists to speak to student groups may write to the Educational Relations Section, Public Relations Staff, General Motors Technical Center, P. O. Box 177, North End Station, Detroit 2, Michigan.

The following GM personnel made recent technical presentations.

Automotive Engineering

Harry F. Barr, chief engineer, Chevrolet Motor Division, before the American Automotive Leasing Association, Chicago, title: The Chevrolet Corvair and the Automotive Leasing Industry.

Clare D. Harrington, administrative engineer, Oldsmobile Division, before Masonic Lodge, Alma, Michigan, title: Horsepower for Safety.

J. C. Kindermann, spark plug engineer, AC Spark Plug Division, before Flint, Michigan, Power Squadron, title: Operation and Care of Marine Engine Spark Plugs.

Craig Marks, assistant engineer in charge, Power Development Group, GM Engineering Staff, and **Peter Kyropoulos**, technical director, GM Styling Staff, before the 34th annual meeting of the California Natural Gasoline Association, Pasadena, title: The Potential of Unconventional Powerplants for Vehicle Propulsion.

Donald P. Marquis, assistant chief engineer, Saginaw Steering Gear Division, before the Saginaw, Michigan, section, Society of Automotive Engineers, title: Future High Angle Universal Joints.

Les Murray, staff engineer, Cadillac Motor Car Division, before the Service Manager Club, Charlotte, North Carolina, district, title: Field Service Problems Related to Electrical and Accessory Components.

Max M. Roensch, assistant chief engineer, Experimental Test Section, Chevrolet Motor Division, before the European Mobil Oil Company chief engineers meetings, Copenhagen, Denmark, and Lisbon, Portugal, title: Chevrolet's New Engineering Laboratory.

Sheldon T. Smith, supervisor, engineering services, Delco Appliance Division, before the LeRoy, New York, Rotary Club, title: Automotive Power Assists.

Russell E. Stebar, Robert L. Everett, research engineers, and **Warren M. Weise**, senior research engineer, GM Research Laboratories, before the S.A.E. international west coast meeting, Vancouver, British Columbia, Canada, title: Engine Rumble—A Barrier to High Compression Ratios?

From the GM Engineering Staff: **B. J. Mitchell**, assistant engineer in charge, Power Development Group, before Oklahoma A and M Third Annual Industrial Engines Short Course, Norman, title: A Look Into the Future of Reciprocating Engines, and **Howard K. Gandelot**, vehicle safety engineer, before the Genesee County Bar Association, Flint, Michigan, title: Building Safety Into Today's Cars.

From Buick Motor Division: **Philip C. Bowser**, director, reasearch and development, before students of General Motors Institute, Flint, title: Chassis Design—Its Objectives and Limitations, and **William Edgley**, chief draftsman, before students of Southwestern High School, Flint, title: Automotive Design.

From Detroit Diesel Engine Division: **Kenneth L. Hulsing**, staff engineer, before students of Pennsylvania State University enrolled in a course in maintenance of commercial vehicles, title: The Diesel Engine, and **Frank W. Sinks**, application engineer, before a regional S.A.E. meeting, St. Louis, title: Application of Diesel Engines.

From GMC Truck and Coach Division: **N. E. Wheeler**, welding supervisor, before the American Welding Society, Detroit, title: Fabrication of a Lightweight Truck Frame; **Donald J. LaBelle**, truck engineer, before the Central Motor Freight Association, Chicago, title: Trucks for Tomorrow; and **W. E. Whitmer**, Diesel project engineer, and **W. P. Strong**, coach engineer, before the American Transit Association, Minneapolis, titles: Factors Affecting Fuel Economy and New Transit Coach Construction, respectively.

From the GM Proving Ground: **Louis C. Lundstrom**, director, before the regional committee on design, American Association of State Highway Officials, Denver; the Ohio Highway Department, Columbus; and the annual meeting of the American Association of State Highway Officials, Boston, title: Roadside Hazards; **Kenneth A. Stonex**, assistant director, before the 10th National Conference of the American Standards Association, Detroit, title: Engineering and Regulatory Standards in Highway Construction and Traffic Control; and **Thomas M. Fisher**, administrative engineer, before the Second Annual Asphalt Pavers Conference, Milwaukee, title: Elimination of Roadside Hazards.

Before the Sixth Annual Automotive Air Conditioning Forum, Texas S.A.E. section, Dallas: **William H. Jackson**, supervising engineer, Harrison Radiator Division, title: Design and Test of an Automotive Air Conditioning System, and **William K. Steinhagen**, assistant engineer in charge, Power Development Engineering Staff, title: Automotive Air Group, GM Conditioning Compressors—Past, Present, and Future.

Fisher Body Division engineers who made presentations at the 14th Annual Technical Convention, American Society

of Body Engineers, Detroit, included **Arthur S. Bassette**, assistant engineer in charge, Fisher Body Proving Grounds Activity, title: Electronic Testing of Body Mechanisms, and **William E. Sehn**, engineer in charge, Product Evaluation Department, chairman of the materials and techniques session.

Rochester Products Division personnel who showed the Rochester Products film "Joe Period and the Carburetor" and discussed carburetor engineering included: **Theodore H. Redman**, senior project engineer, before the Brockport, New York, Lion's Club; **James F. Hughes**, director of public relations, before the LeRoy, New York, Rotary Club; and **Jack R. Barr**, supervisor, safety and suggestions, before the Webster, New York, Lion's Club.

Before the S.A.E. National Transportation, Diesel Engine, and Fuels and Lubricants Meeting, Chicago: **Theodore W. Selby**, senior research chemist, GM Research Laboratories, title: Automatic Transmission Fluid Viscosity at Low Temperature and its Effect on Transmission Performance; **George J. Mach**, assistant staff engineer, Chevrolet Motor Division, title: Chevrolet's Approach to Truck Tire and Wheel Problems; and **Harold O. Flynn**, assistant chief engineer, truck body and chassis design, Chevrolet Motor Division, title: The Design of a Truck for Better Ride and Handling.

Bearings

J. G. Turnbull, bearing sales engineer, Allison Division, before the Northwest Carmens' Association meeting, Minneapolis, title: The Story of the Allison KAR-GO Bearing.

From New Departure Division: **Robert B. Walker**, project engineer, before engineers of the Century Electric Company, St. Louis, title: Effect of Ball Bearings on Airborne Noise of Electric Motors; **Thomas W. Bakewell**, supervisor, general bearing application, before engineers of the Gorman Rupp Corporation, Mansfield, Ohio, and the F. E. Meyers Corporation, Ashland, Ohio, title: Selection and Application of Ball Bearings; and **A. H. Kelso**, manager, instrument bearing development and contract, before the Sandusky, Ohio, Industrial Management Club, title: Bearings and Missiles.

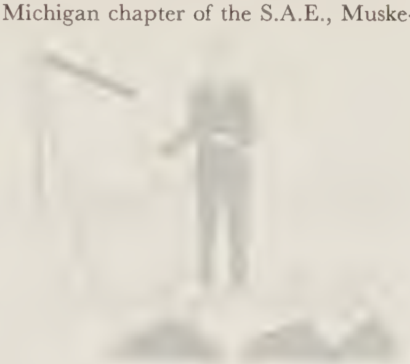
Richard C. Drutowski, supervisor, Mechanical Development Department, GM Research Laboratories, and E. B. Mikus, no longer with GM, before the A.S.L.E.-A.S.M.E. Lubrication Conference, New York City, title: The Effect of Ball Bearing Steel Structure on Rolling Friction and Contact Plastic Deformation.

Computers

Dale E. Benedict, project engineer, Delco Products Division, before the Dayton, Ohio, section of the American Institute of Electrical Engineers, title: Digital Computer Usage at Delco Products.

K. L. Nielsen, chief, operations, Analysis Section, Allison Division, before the Annual Indiana State Teachers Convention, Indianapolis, title: Modern Mathematics and the Junior High School, and before the 75th Annual Meeting of the Indiana Academy of Science, Butler University, Indianapolis, title: Computers and the Collegiate Mathematics Program.

From the GM Research Laboratories: **Edwin L. Jacks**, supervisor, Special Problems Department, before the western Michigan chapter of the S.A.E., Muske-



gon, title: The Digital Computer—Tool for Engineers, and **John L. Harned**, senior research engineer, and **D. C. Fosth**, research engineer, Electronics-Instrumentation Department, before the 14th Annual I.S.A. Instrumentation-Automation Conference, Chicago, title: Analog Computer Data Recovery Technique.

Electrical Engineering

Stanley F. Newman, senior electrical engineer, GM Process Development Staff, before the A.I.E.E., Chicago, title: Electrical Standards in Industry.

B. H. Hefner, electrical engineer, Electro-Motive Division, before the district meeting of the A.S.M.E., Philadelphia, title: Merits of EMD Generating Power for De-centralized Utility Application.

Bernard J. Pleiss, project engineer, Delco Products Division, before the Dayton, Ohio, section of A.I.E.E., title: Synthetic Insulation for Hermetic Motors.

Before the American Association of State Highway Officials, Electronics Committee, Boston: **J. R. Atkinson**, senior project engineer, Delco Radio Division, title: Description of Prototype Hy-Com Equipment, and **Clark E. Quinn**, senior research engineer, GM Research Laboratories, title: The GM Highway Communication System—Hy-Com.

From Delco Radio Division: **F. L. Hughes**, supervisor, field services, before the Kokomo, Indiana, Rotary Club, title: Transistors—Today and Tomorrow; **R. H. Wright**, service engineer, before students of Los Angeles Trade Technical College, title: Transistors and Transistor Products; **G. M. Ford**, resident applications engineer, west coast office, before the A.I.E.E., Los Angeles, title: Ten Pitfalls in Power Transistor Applications, **J. S. Schaffner**, manager, applications and evaluations, before the Southwestern Industrial Electronics Engineering Seminar, Houston, and engineering personnel of Bendix Aviation, Kansas City, Kansas, title: Application of Power Transistors in Square Wave Oscillators; **J. R. Atkinson**, senior project engineer, before the American Bridge Tunnel and Turnpike Association, Kansas City, Kansas, title: Hy-Com Zone Communications Control; and **W. C. Caldwell**, service engineer, before the mid-west convention of the American Radio Relay League, St. Louis, title: The Transistor, How Does It Work, before students of Northwestern High School, Kokomo, title: Getting Started in Radio, and before students of the American Television and Electronics School, Muncie, Indiana, title: Transistors—Fundamentals and Circuits.

Foundry

C. E. Drury, plant manager, Danville, Illinois, Plant, Central Foundry Division, before the St. Louis chapter of the American Foundrymen's Society, title: Pouring Effect on Scrap.

David C. Salatin, staff engineer, GM Process Development Staff, before the Society of Die Casting Engineers, Ander-

son, Indiana; title: Is Die Casting the Answer? Why Isn't It?

Robert M. Critchfield, formerly vice president in charge of GM Process Development Staff, now retired, before the Detroit chapter of A.F.S., title: Developments in Casting Processes.

Robert F. Thomson, head, Metallurgical Engineering Department, GM Research Laboratories, before the Purdue University Casting Conference, Lafayette, Indiana, title: Applications for Aluminum Castings in the Automotive Industry.

Guided Missiles

Engineering personnel of AC Spark Plug Division's Milwaukee, Wisconsin, plant who made recent presentations on guided missiles and inertial guidance systems include: **Hans Hauser**, supervisor, MACE project administration, before the Milwaukee Rotary Club, title: Missile Guidance; **Robert Ahrens**, program director, MACE engineering, before the West Allis, Wisconsin, Kiwanis Club, title: Missile Guidance; **Harold Raninen**, chief engineer, MACE project, before the aviation section, A.S.M.E., Chicago, title: Missiles; **Joseph Shea**, director, advanced systems research and development, before the annual Industry-Clergy Day of the Milwaukee Kiwanis Club, title: Missiles and Religion in the Space Age; **B. R. Spink**, Thor engineer, **Wilhelm Steffe**, Titan engineer, and **Frank Knopf**, Titan engineer, before the Fourth Annual Symposium on Ballistic Missile and Space Technology, University of California, Los Angeles, titles: A Practical Solution to the Arcing Problem at High Altitude, Design Factors and Performance Characteristics of a 500-VA Static Inverter, and Investigation of Capability of Precision Calibration of Pendulous Gyroscope Accelerometers on Centrifuge, respectively; and **A. J. Italiano**, head, Thor project office, before the Milwaukee Optimist Club, title: Missiles From Here to There, and before the West Allis, Wisconsin, Rotary Club and the Circle K Club of Carroll College, Waukesha, Wisconsin, title: Missiles.

H. L. Karsch, project manager, advanced weapons systems, Allison Division, before the General Motors Institute Alumni Association, Indianapolis, title: Early Foreign Rocket Development.

Charles N. Hay, supervisor, contract research, New Departure Division, before the Sandusky, Ohio, Exchange Club, title: New Departure's Contribution to Missile Guidance.

L. E. A. Batz, scientific gyroscope advisor, AC Spark Plug Division, before a joint meeting of the New London, Connecticut, section of the A.S.M.E. and personnel of the Electric Boat Company, title: Gyroscopic and Inertial Navigation, and before the Eagle Scout Recognition Banquet, Lansing, Michigan, title: The Gyroscope and its Uses.

Hydraulics

Richard Handwerker, instructor General Motors Institute, before American Society of Lubricating Engineers, Detroit; title: Basic Hydraulics.

Before the National Conference on Industrial Hydraulics, Chicago: **F. L. Mackin**, administrative chairman, General Motors Institute, title: Industrial Hydraulics; **Robert M. Van House**, supervisor, Engineering Mechanics Department, GM Research Laboratories, title: Hydraulics in Future Highway Transportation; and **Ward F. Diehl**, senior hydraulics engineer, GM Process Development Staff, title: Air Power and Control in Industry.

Instrumentation

Albert F. Welch, head, Electronics-Instrumentation Department, GM Research Laboratories, before the Gordon Research Conference on Instrumentation, Colby Junior College, New London, New Hampshire, title: Instrumentation in Automotive Research.

R. D. Tyler, superintendent, electronics and parts tests, Aircraft Engines Operations, Allison Division, before the Central Indiana Section of the Society for Experimental Stress Analysis, Indianapolis, title: Some Unique Vibration and Dynamic Stress Instrumentation Techniques Used at Allison Division.

Paul C. Skeels, head, Experimental Engineering Department, GM Proving Ground, before the Columbus, Ohio, section of the Instrument Society of America, title: Proving Ground Instrumentation.

Before the 14th Annual I.S.A. Instrumentation-Automation Conference, Chicago: **John R. Hunsberger**, research engineer, GM Research Laboratories, title: Firebird III Instrumentation; **A. H. Kelly**, head, Engineering Test Department, GM Proving Ground, title: GM Proving Ground and Photographic Instruments; **William Shepherd**, senior project engineer, Frigidaire Division, title: Acoustic Instrumentation as Applied to Noise and Vibration Analysis of Household Appliances, and **Albert F. Welch**, head, and **Russell V. Fisher**, general supervisor, Electronics Instrumentation Department, GM Research Laboratories, title: Organizing Today to Solve the Instrumentation Problems of Today and Tomorrow.

Manufacturing

C. R. Gillette, manager, research chemistry, New Departure Division, before the Esso New England District Sales Conference, Hartford, Connecticut, title: Rust Prevention Practice in Manufacturing.

Robert Bentley, senior engineer, GM Process Development Staff, before the Saginaw Valley chapter, American Society of Tool Engineers, Flint, title: Electrical Discharge Machining.

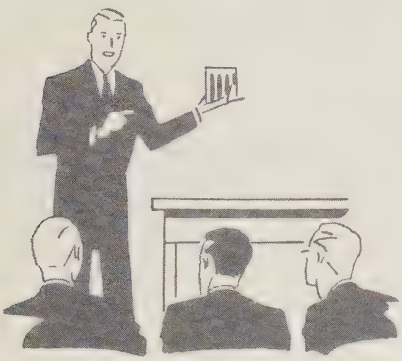
Robert B. Colten, staff engineer, GM Process Development Staff, before the 47th National Safety Congress, Chicago, title: Safety Inspection of Machinery by use of Ultrasonics.

Walter G. Maher, director of reliability, Rochester Products Division, before the Rochester, New York, chapter, American Society of Quality Control, panel member on discussion of reliability.

From Allison Division: **C. R. McDowell**, statistician inspection, before the Indianapolis, Indiana, chapter, American Institute of Industrial Engineers, title: Work Sampling; **C. B. Butts**, test engineer, before the Society of Non-Destructive Testing, Chicago, title: Automated Ultrasonic Inspection of Jet Engine Rotor Forgings; **A. R. Townsend**, manager, reliability, before the third Navy-Industry Conference on Aeronautical Material Reliability, Virginia Beach, Virginia, title: Measuring the Merit of Field Service Changes by a Reliability Technique; and **D. F. Flanders**, department head, Administrative Services, before the Industry-Military Quality Control Management Symposium, title: Allison's Quality Control Management Team.

N. J. Gebhart, general manager, Moraine Products Division, before Western Teachers Association of Ohio, Dayton, title: Metal Powder Production Techniques.

Metallurgy



Harold F. Sprague, supervisor, general metallurgy, New Departure Division, before Connecticut Industrial Gas Workshop, Chesire, Connecticut, title: Ferrous Metal Heat Treating and the Application of Prepared Atmospheres.

James C. Holzwarth, supervisor, Metallurgical Engineering Department, GM Research Laboratories, before the combined meeting of the Ohio Valley chapter of the National Association of Corrosion Engineers and the Louisville, Kentucky, chapter of the American Society for Metals, Louisville, title: Automotive Corrosion Test Methods and Facilities Used by General Motors.

Before the American Society for Metals' National Metals Congress, Chicago: **D. K. Hanink**, chief metallurgist, Allison Division, title: Steel in Solid Fuel Rocket Motor Cases; and **Thomas J. Hughel**, supervisor, Metallurgical Engineering Department, GM Research Laboratories, E. B. Mikus, no longer with GM, and **J. M. Gerty** and **A. C. Knudsen**, AC Spark Plug Division, title: The Dimensional Stability of a Precision Ball Bearing Material.

Miscellaneous

George P. Ransom, section engineer, GM Engineering Staff, before the American Society of Heating and Air Conditioning Engineers, Baton Rouge, Louisiana, title: The Axial Type Refrigeration Compressor.

From the GM Research Laboratories: **Alvin E. Roberts**, senior project engineer, before the national meeting of the Institution of Sanitation Management, New York City, title: Relamping and Cleaning of Lighting Fixtures; **Harold A. Kahler**, liaison engineer, before the Automatic Car Wash Association, International, Detroit, title: Care of Chromium Plated Trim and Stainless Steel Trim on Automobiles; and **Lloyd L. Withrow**, head, Fuels and Lubricants Department, before a meeting celebrating the oil industry's centennial year attended by oil men's groups of Michigan and S.A.E. engineers, Detroit, title: What Will Happen if the Free World Loses the Middle East Oil?

From the GM Proving Ground: **A. H. Kelly**, head, Engineering Test Department, before the southeastern Michigan branch of the American Meteorological Society, Ann Arbor, title: GM Proving Ground and the Weather, and **Richard O. Painter**, head, Engineering Photographic Department, before the University of Wisconsin Extension Division Engineering Institute, Madison, title: Applications of High Speed Photography in the Automotive and Allied Industries.

From AC Spark Plug Division: **Karl Schwartzwalder**, director of research, before the American Ceramic Society, Pittsburgh section, Mellon Institute, title: Alumina Ceramics, and before the American Chemical Society, Flint sub-section, title: Technological Problems of Ceramic Insulators; **James C. White**, senior procedure writer, before the Systems for Management Course, sponsored by Marquette University in cooperation with the Systems and Procedures Association of America, Milwaukee, title: The Nature of Systems for Management; and **Ralph O. Helgeby**, staff engineer, before the Flint Optimist Club, title: European Travel.

Before the American Standards Association, Detroit: **Clyde L. Fanning**, assistant chairman, engineering shops, General Motors Institute, title: The Unique Position of an Engineering Society in Standards, and **R. H. Bertsche**, electrical engineer, GMC Truck and Coach Division, title: Role of Communication in International Standards.

From Delco Radio Division: **Martin J. Caserio**, general manager, before supervisors of Central Indiana Bell Telephone Company, Kokomo, title: Secrets of Job Success, and **G. E Dieterly**, senior process engineer, before the Metropolitan Kiwanis Club, Kokomo, title: Plastics.

From General Motors Institute: **Harold M. Dent**, administrative chairman, cooperative engineering program, before Chemistry Section, Michigan Education Society, Detroit, title: Engineering at GMI; **Charles J. Sahrbeck, Jr.**, administrative chairman, before Lions Club, Flushing, Michigan, title: Management Training in American Industry; and **Eugene Sullivan**, program supervisor, before Indiana Dairymen's Association, Indianapolis, title: The Supervisor Accepting His Responsibility on the Management Team.

Henry S. Smith, administrative engineer, Saginaw Steering Gear Division, before Lions Club, Alma, Michigan, title: The Saginaw Story.

Charles C. Wardell, advertising manager, Hyatt Bearings Division, before the Sales Engineering Conferences, Newark College of Engineering, Newark, New Jersey, title: How to Develop and Use Product Information.

From Allison Division: **H. B. First**, manager, Service Department, Aircraft Engines Operations, before Air Transport Association, New Orleans, title: Overhaul and Development of the Allison 501-D13 Prop-jet Engine, and **Harold Williams**, supervisor, turbo-prop engineering training, before the Kokomo, Indiana, Rotary Club; title: Lockheed Electra Propulsion System.

Research

Personnel from the GM Research Laboratories who made recent presentations included:

H. M. Bendler, research associate, before the Michigan Electron Microscopy Forum, Warren, title: Contrast in Electron Microscope Images.

Robert R. Bockemuehl, and **Walter E. Sargeant**, senior research engineers, before Conference on Magnetism and Magnetic Materials (A.I.E.E., I.R.E., O.N.R., A.P.S., A.I.M.E.), national meeting, Detroit, title: A Practical Hysteresigraph.

Lawrence D. Dyer, senior research physical chemist, before the American Crystallographic Association, 1959 annual meeting of Michigan Affiliate, Michigan State University, East Lansing, title: Rolling Friction on Single Crystals of AgCl and Copper.

Anthony Foderaro, senior research scientist, before a reactor training course for British engineers and scientists, Pittsburgh, title: Nuclear Submarine Biological and Thermal Shielding.

Robert C. Frank, senior research physicist, before the Society for Applied Spectroscopy, Milwaukee, title: Mass Spectroscopy in Research.

David L. Fry, supervisor, Physics Department, before the A.S.T.M. symposium on emission spectroscopy, national meeting, San Francisco, title: Communications in Spectrochemical Analysis.

Charles W. Gadd, supervisor, Special Problems Department, before the national meeting of the Society for Experimental Stress Analysis, Detroit, title: Experimental Mechanics in Design.

Farno L. Green, senior research physicist, before the Radiological Society, Memphis, Tennessee, title: Are Radioactive Sources Which Emit Low Energy X- and Gamma-Rays Potentially Useful in Medicine?

William L. Grube, assistant head, Physics Department, before the national meeting of the Electron Microscope Society of America, Columbus, Ohio, title: Preparation and Replication of Narrow Cross Sections for Electron Metallography.

F. Earl Heffner, senior research engineer, before the A.S.L.E.-A.S.M.E. Lubrication Conference, national meeting, New York City, title: A General Method for Correlating Labyrinth Seal Leak Rate Data.

Robert Herman, head, Theoretical Physics Department; J. W. Follin, Jr., Johns Hopkins University; and R. A. Alpher, General Electric Company; before the American Physical Society, national meeting, Los Alamos, title: Light Element Formation During Early Stages of the Expanding Universe.

Marvin W. Jackson, research engineer, and W. B. Heaton, no longer with GM, before a regional meeting of the Association of Analytical Chemists, 7th Detroit Anachem Conference, title: Hydrocarbons in Exhaust Gas, Non-Dispersive Infrared Versus Gas Chromatographic Measurements.

F. E. Jamerson, senior nuclear physicist, before the American Rocket Society, Detroit, title: Nuclear Reactors in Space.

Robert W. Lee, research physicist, and **Robert C. Frank**, senior research physicist, before the international meeting of the Societe Francaise de Metallurgie (presented in absentia), Paris, title: Mass Spectrometer Studies of the Diffusion of Hydrogen in Iron.

Richard E. Marburger, senior research physicist, before the General Electric Diffraction School, Milwaukee, title: X-ray Residual Stress Measurements on Hardened Steels.

Thomas O. Morgan, research chemist, before the General Motors Summer Program for High School Science Teachers, Warren, Michigan, title: Chromatography.

Gerald M. Rassweiler, head, Physics Department, before the national meeting of the American Institute of Physics, Harriman, New York; title: The Role of the Physicist in the Automotive Industry.

Bernard J. Riley and **Seward E. Beacom**, senior research chemists, before the Third Industrial Nuclear Technology Conference, Chicago, title: Radiotracer Techniques Useful in Electrodeposition Studies.

Thomas P. Schreiber, senior research physicist, and **Richard F. Majkowski**, research physicist, before Sixth Ottawa Symposium on Applied Spectroscopy, Ontario, Canada, title: The Minimizing of Matrix Effects in the Emission Spectrographic Analysis of Tool Steels.

James D. Thomas, senior research engineer, before the American Electroplaters Society, conference of the midwest regional council, Rockford, Illinois, title: The Copper Chloride Modification of the Acetic Acid Salt Spray Test.

William A. Turunen, head, Engineering Development Department, before the Cleveland, Ohio, section of the S.A.E., title: The GT-305 Regenerative Gas Turbine Engine.

Philip Weiss, head, Polymers Department, before European universities at London, Louvain, Brussels, Paris, Zurich, Milan, Turin, Rome, and Vienna, titles: Vinyl Type Graft Copolymers and High Temperature Polymers.

Before the national meeting of the American Chemical Society, Atlantic City, New Jersey: **Bernard J. Riley** and **Seward E. Beacom**, senior research chemists, title: Some Radiotracer Techniques Applicable to Studies of Electrodeposition Phenomenon, and **Seward E. Beacom**, title: The Photochemical Reactions of Some Transition Metal Complexes.

Science and Education

Craig Marks, assistant engineer in charge, Power Development Group, GM Engineering Staff, before Lamphere Public Schools Science Seminar Program Madison Heights, Michigan, title: The Scientific Approach.

From the GM Research Laboratories: **Lawrence R. Hafstad**, vice president in charge, at dedication of Linke Hall, Linfield Research Institute, McMinnville, Oregon, title: Science and Opportunity, and **Seward E. Beacom**, senior research chemist, before the Macomb County Science and Mathematics Teachers Association, Mt. Clemens, Michigan, title: Some Frontiers in Science Education, and before the GM Summer Program for High School Science Teachers, Warren, Michigan, title: The Scientist and Frontiers of Creativity.

From AC Spark Plug Division: **Willard E. Hauth**, staff scientist, before students of Southwestern High School, Flint, title: Science Research; **Lavern Maurand**, research metallurgist, before students of Central High School and Southwestern High School, Flint, title: Metallurgical Engineering; and **Robert W. Smith**, staff scientist, before students of Central High School and Northern High School, Flint, title: Science Research, and before the Swartz Creek, Wisconsin, Kiwanis Club, title: Science Education for Non-Scientists.

Thomas B. Watson, supervisor, drafting training, Buick Motor Division, before students of Central High School and Northern High School, Flint, title: Automotive Design and Drafting as a Career.

Joseph M. Rodgers, general supervisor, and **Charles R. Ginn**, supervisor, Electrical Laboratory, Delco Products Division, before a Careers Day program for Dayton high schools, title: Engineering a Living.

GM Styling Staff Aquaints Educators With Automotive Design Problems

A NEW method of presenting information about General Motors activities to educators was introduced during the GM Styling Staff's program for the 1959 GM Conference for Engineering and Science Educators*. This part of the Conference, called the *Educator Design Workshop*, gave the 28 visiting educators the opportunity to analyze and solve automotive styling problems by designing a hypothetical, new car model.

To show the educators how it develops a basic automobile styling concept from a given set of data, the Styling Staff conducted the Educator Design Workshop in three phases:

(a) Compiling a set of car specifications

(b) Reaching basic automotive styling decisions during a design workshop

(c) Refining the design theme and preparing full-size drawings and renderings.

The Specifications

Prior to the Conference, each educator completed a questionnaire that listed the dimensions, appearance, performance, and general characteristics which he preferred in an automobile. This information was compiled into one set of specifications by members of the Styling Staff and presented to the educators at the beginning of the workshop session.

The Workshop

Moderator for the workshop was Dr. Peter Kyropoulos, technical director, who was assisted by Donald Hoagg, chief designer, Design Development Studio; Stefan Habsburg, assistant chief designer, Research Studio; and Judson Holcombe, research engineer; all of the GM Styling Staff. They demonstrated some of the techniques used in designing new car models and guided the educators during their Conference experience as automotive stylists.

Using data from the approved set of specifications for the car, the proposed width, length, height, ground clearance, and wheelbase were plotted on a large board by GM stylists. Next, paper cutouts representing the gasoline tank, headlights, wheels, spare tire, engine, and battery were pinned on the board in positions selected by the educators.

Two life-size cardboard "dummies," based on the dimensions of the average and above average American, along with cutouts representing the front and rear seats, then were shifted on the board to determine a seat placement that provided for sufficient leg and head room for the driver and passengers. Various door contours were investigated and the car's preliminary weight distribution was calculated. Since the windshield is an important safety and visibility factor, its size and location also received special consideration during the workshop.

The educators then planned the car's exterior styling by using a technique popular with GM stylists. Various colored strips of yarn were pinned on the board to form a general outline around the dimensions and mechanical components which already were in place. The yarn was shifted to conform to any design theme the group wanted to investigate.

After the exterior appearance had been discussed, two of the educators, George A. Jergenson, vice president and head of the Industrial Design Department at The Art Center School, Los Angeles, and Jay Doblin, director of the Institute of Design at the Illinois Institute of Technology, Chicago, were assigned to complete the project on their three-day field assignment to the Styling Staff during the second week of the Conference.

The Final Design

Early in the final phase of the Educator Design Workshop, Mr. Jergenson and Mr. Doblin decided to design separate cars, representing their individual design concepts; yet both designs corresponded to the design specifications and recommendations of the educator group. To assist in the preparation of artwork necessary in developing the new car designs, each educator was assigned a team of student designers who were summer employes at the Styling Staff. Functioning in the capacity of a design studio director, the two educators supervised the students' work and contributed design sketches of their own.

Numerous small-scale sketches were prepared by each design team to test the educators' ideas on paper. From these sketches, a basic styling theme was developed and was transformed into full-size drawings and renderings. At the final session of the Educator Conference, attended by the entire educator group and by GM participants, a cardboard cutout of Mr. Jergenson's model was displayed along with a rendering of Mr. Doblin's design (Fig. 1).

The Educator Design Workshop enabled the visiting educators to experience an accelerated version of the step by step procedure involved in automotive styling. This included an apprecia-

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*See page 42, October-November-December 1959 issue of the GENERAL MOTORS ENGINEERING JOURNAL for a summary of the complete conference program.

tion for the close cooperation needed between stylists, product engineers, and manufacturing engineers to produce a satisfactory design of the automobile.

The 1959 Conference for Engineering and Science Educators was sponsored jointly by the Engineering, Research, Process Development, and Styling Staffs. It was the eighth conference of this type held. It was designed to give GM management a better understanding of current problems in educating future engineers and scientists and to give educators firsthand information on how engineering and science are used in General Motors.

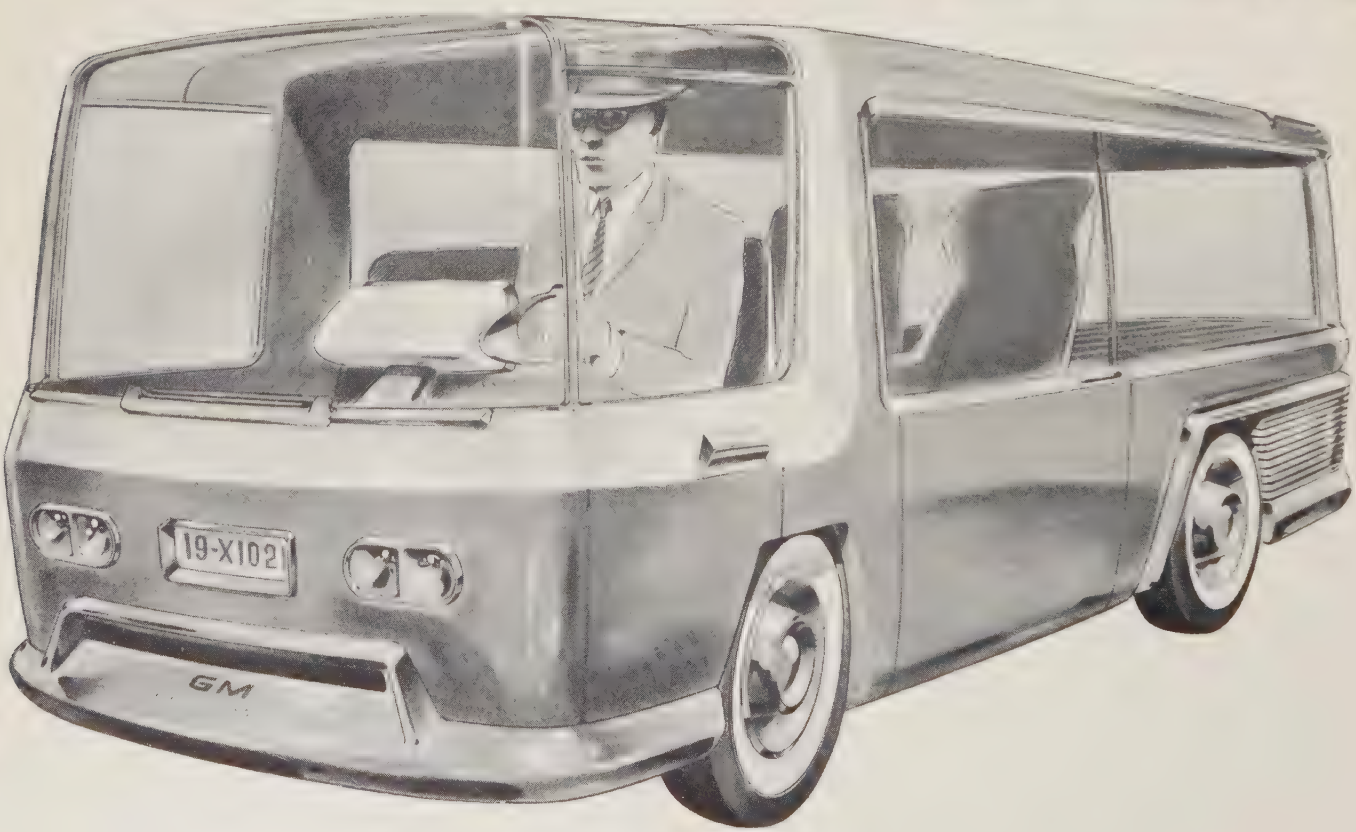


Fig. 1—The different design approaches used by the two educator-stylists in complying with the recommendations of the educator group are shown here. George A. Jergenson (The Art Center School) describes his model, represented by a full-size cardboard cutout (top), to the educators and GM participants at the GM Technical Center on the last day of the 1959 Educator Conference. Jay Doblin (Illinois Institute of Technology) is shown completing a preliminary sketch of his car (center) during his three-day field assignment to the Styling Staff. A full-size drawing of Mr. Doblin's car (bottom) illustrates his functional approach to the problem. Some of the features suggested by the educators and designed into the two models were simple lines, economical operation, good vision, comfort, and safety.

Solution to the Previous General Motors Institute Classroom Problem:

Determine the Entropy Values for Low Pressure, Superheated Water Vapor

Faculty Member in Charge:
ELWOOD K. HARRIS

G.M.I. Cooperative Student:
ROBERT D. FLETCHER
Electro-Motive Division

The thermodynamic properties of water vapor are probably better known than those of any other substance. The numerous engineering problems that arise when water and water vapor must be considered require that knowledge of its behavior be expanded whenever the need arises. Presented here is the solution to a problem which illustrates a case in point. The problem, presented in the October-November-December 1959 issue of the GENERAL MOTORS ENGINEERING JOURNAL, was based on a need to establish more information dealing with superheated water vapor at low pressures and elevated temperatures. Finding this information required the application of thermodynamic principles to obtain data for plotting an extension to cover that portion of a temperature-entropy diagram dealing with low pressures and elevated temperatures.

A typical application of applied thermodynamics

To study the effect of ambient temperature and pressure conditions on internal combustion engine performance, data were needed which would illustrate graphically property relations for superheated water vapor at low pressures and elevated temperatures. To determine these values a temperature-entropy (T-S) diagram had to be extended to include the temperature and pressure range under study.

The first step is to derive an equation to calculate the change in entropy. A number of reversible, non-flow thermodynamic processes* could be chosen as the basis for deriving this equation—for example, an isothermal process, a constant pressure process, or a constant volume process. For this solution, an isothermal, non-flow process was chosen because of its linear plot on the T-S diagram.

Experimentation has shown that the properties of superheated vapor in a range of low pressures and high temperatures can be predicted within reasonable engineering accuracy by means of the perfect gas laws[1]. For the purpose of the solution, therefore, the perfect gas laws were used.

The equation for expressing the change in entropy ΔS can be derived as follows.

The differential form of the energy equation for a non-flow process, when neglecting such effects as gravitational effects, magnetism, and electricity, may be expressed as

$$dQ = dU + \frac{dWk}{J} \qquad (1)$$

where

dQ = heat added or subtracted
dU = change in internal energy
dWk = work done on or by the working fluid
J = a constant = 778 ft-lb per Btu.

Presented here is the solution to a problem resulting from a G.M.I. engine design classroom discussion question, "What is the effect of atmospheric vapor on the charge inducted into the cylinder of an internal combustion engine?" To answer the question required the application of thermodynamic principles to establish data useful for plotting an extension to a temperature-entropy diagram to cover the low pressures and elevated temperatures usually encountered in internal combustion engine operation.

From the definition for the change of entropy for a reversible process, the following equation results.

$$dQ = TdS \qquad (2)$$

where

dQ = element of reversible heat
T = absolute temperature (°R)
dS = change in entropy.

The change in internal energy dU can be expressed as

$$dU = Wc_v dT \qquad (3)$$

where

W = weight
c_v = specific heat at constant volume.

Also, the work done is equal to

$$dWk = PdV \qquad (4)$$

where

P = absolute pressure (psfa)
dV = change in volume (cu ft).

Substituting terms from equations (3) and (4) into equations (1) and (2) and solving for dS gives

$$TdS = dQ = Wc_v dT + \frac{PdV}{J}$$

$$dS = \frac{Wc_v dT}{T} + \frac{PdV}{JT} \qquad (5)$$

$$P = \frac{WRT}{V}.$$

*In the statement to the problem, which appeared on page 39 of the October-November-December 1959 issue, three initial assumptions were made. These assumptions should not have been included.

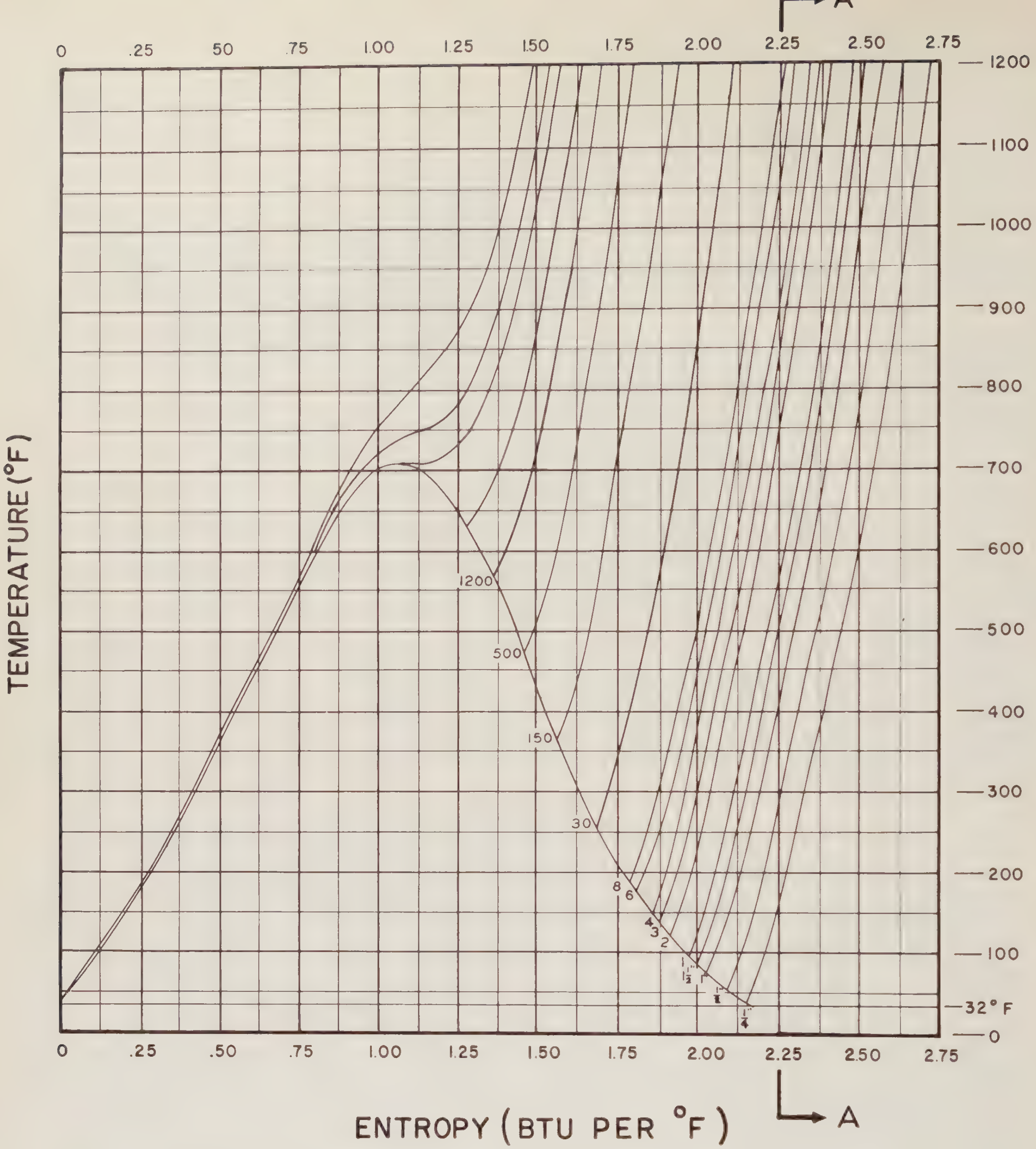


Fig. 1—Shown here is the extension made to an existing temperature-entropy diagram (in the direction indicated by arrows *A-A*) to provide information on low pressure, superheated water vapor for a temperature range of from 0°F to 1,200°F.

For an isothermal process and a perfect gas,

$$dT = 0$$

$$dU = 0.$$

Equation (5), therefore, becomes

$$dS = \frac{PdV}{JT}. \qquad (6)$$

Substituting the equality of P into equation (6), gives the following value for dS:

$$dS = \frac{WRTdV}{TJV} = \frac{WRdV}{JV}. \qquad (7)$$

Therefore,

$$\Delta S = \frac{WR}{J} \int_{V_1}^{V_2} \frac{dV}{V}$$

$$\Delta S = \frac{WR}{J} \ln V \Big]_{V_1}^{V_2} = \frac{WR}{J} \ln\left(\frac{V_2}{V_1}\right) \qquad (8)$$

From Boyle's Law,

$$P_1V_1 = P_2V_2.$$

Substituting into equation (8),

$$S_2 - S_1 = \frac{WR}{J} \ln \left(\frac{P_1}{P_2}\right) \qquad (9)$$

where

S_1 = entropy for a given temperature and pressure on the existing T-S diagram and in tables

S_2 = calculated entropy value at pressure P_2 and given temperature

P_1 = pressure which is taken as a base on the existing T-S diagram and in tables

P_2 = final pressure

W = weight

R = 85.8 ft-lb per lb°R = gas constant for water vapor

J = 778 ft-lb per Btu = mechanical equivalent of heat.

Examination of equation (9) shows that the change in entropy will be dependent on the pressure ratio because the remaining expressions are constants. Therefore, the change in entropy for a given pressure ratio will be the same for all temperatures in the range being considered and under the assumed conditions. This permits an extension of the constant low pressure curves (1 psia to ¼ inch of mercury) to 1,200°F provided they are held parallel to the 1 psia curve. Extension of constant pressure curves between 10 psia and 1 psia is made possible by selecting temperature and entropy values from superheated vapor tables[2].

To determine the validity of assumptions being made, test points have been calculated and compared to published values of thermodynamic properties of steam[2].

The following two test point calculations illustrate the procedure used. Similar calculations are used for various pressure ratios and checked against known quantities.

Test Point 1

Assumed:

$$P_1 = 2 \text{ psia}$$

$$P_2 = 1 \text{ psia}$$

$$T = 200°\text{F}$$

$$W = 1 \text{ lb}$$

$$S_2 - S_1 = \frac{R}{J} \ln \left(\frac{P_1}{P_2}\right)$$

$$S_2 - S_1 = \frac{85.8}{778} \ln \left(\frac{2}{1}\right)$$

$$S_2 - S_1 = 0.0767$$

$$S_2 = S_1 + \Delta S$$

$$S_2 = 1.9743 + 0.0767$$

$$S_2 = 2.0510$$

Test Point 2

The same pressures are assumed as for Test Point 1. The temperature T, however, is 1,200°F.

$$S_2 = S_1 + \Delta S$$

$$S_2 = 2.3188 + 0.0767$$

$$S_2 = 2.4955.$$

These values of entropy compare within reasonable slide rule accuracy with selected values of 2.0512 at 200°F and 2.4952 at 1,200°F. Per cent error will be reduced by extending lower pressure curves, since vapors behave closer to perfect gases under these conditions.

The T-S diagram (Fig. 1) can be completed by drawing in the constant specific volume, enthalpy, and the degree of superheat lines. This diagram can be helpful in obtaining a mental picture of such things as: what happens to the water vapor during an engine's compression process, what happens to the water vapor in the exhaust gases while cooling at constant pressure, what happens to the water vapor in the exhaust gases when the dew point or saturation temperature is reached and then cooled below this temperature?

These and several other questions can be answered by reasoning with the aid of the T-S diagram.

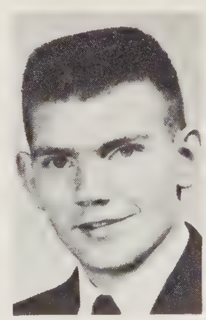
Bibliography

1. Moyer, J. A., Calderwood, J. P., and Potter, A. A., *Elements of Engineering Thermodynamics* (New York: John Wiley and Sons, Inc., 1941), pp. 81-2.
2. Keenan, J. H., and Keyes, F. G., *Thermodynamic Properties of Steam* (New York: John Wiley and Sons, Inc., 1951), Figure 9, Temperature-Entropy Chart.

Acknowledgement

The authors acknowledge the assistance given by the following individuals in the preparation of this problem and solution: Norman Overway, G.M.I. Co-op student, Diesel Equipment Division, and Richard Hughes, G.M.I. Co-op student, Chevrolet Motor Division.

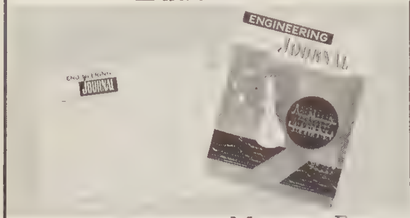
Student Contributor to the General Motors Institute Classroom Problem



ROBERT D. FLETCHER

is a co-op student sponsored by Electro-Motive Division and is presently participating in the G.M.I. Fifth-Year Program which will qualify him for a baccalaureate degree in mechanical engineering when completed. He joined Electro-Motive in 1955. His plant work has included assignments in the equipment engineering and commercial engineering sections of the Product Engineering Department in addition to rotational assignments in the Manufacturing and Manufacturing Service Departments.

About Distribution of the JOURNAL to Educators



The General Motors Engineering Journal is a publication designed primarily for use as supplementary information by college and university educators in the fields of engineering, the physical sciences, mathematics, and industrial design. It is supplied to these educators on a request basis at no charge.

At some institutions there may be educators in the above categories who are not aware that they can receive the Journal on a regular basis. For example, some of these persons might be new members of the faculty. Educators presently on the mailing list of the Journal are invited to inform their colleagues of the availability of this publication.

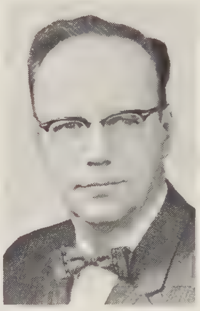
When desired, educators may receive multiple copies of the Journal on a regular basis or for special uses. Copies of most of the back numbers may be obtained on request. The final issue in each Volume of the Journal (designated by calendar year beginning with 1954) contains an index by subject, title of paper, or author.

As announced in the July-August-September 1959 issue, reprints of typical problems in engineering have been arranged in booklet form. A separate booklet contains reprints of the solutions. Both of these items also are available to educators on request at no charge.

Comments from educators always are welcome. In the past, educators' suggestions and advice on subject matter and method of presentation have been helpful in guiding GM scientists and engineers in their preparation of papers for best use in connection with the study of fundamentals.

Contributors to Jan.-Feb.-Mar. 1960 Issue of

ENGINEERING JOURNAL



NEIL W. CARNELL, contributor of "The Flow Box—An Instrument to Measure Fuel Metering Devices for Internal Combustion Engines," is supervisor of test-flow boxes and dynamometers at Rochester Products Division, Rochester, New York.

Mr. Carnell received his engineering education in Detroit, Michigan, and joined the Product Engineering Department of Rochester Products in 1956 as a senior project engineer. One year later he became head of the flow box and dynamometer test area. As supervisor, Mr. Carnell is responsible for the design and development of new test equipment and testing techniques, plus, the procurement of advanced instrumentation for the developmental testing of fuel metering devices. Maintenance of equipment, training of new engineering personnel, scheduling of projects within the test area, and special project assignments also are a part of his responsibilities.

Prior to joining Rochester Products, Mr. Carnell was associated with the Holley Carburetor Company, Detroit, Michigan, from 1939 to 1956 as a project engineer, sales and service engineer, and director of education.

Mr. Carnell is a member of the Society of Automotive Engineers.

GEORGE L. DEMOTT, contributor of "Facts Engineers and Inventors Should Know About the United States Patent Office," served as manager of the Washington-D.C., Office of the General Motors Patent Section from 1952 until his death on August 26, 1959. Mr. DeMott joined General Motors in 1950 as assistant manager of this office. Included in his responsibilities at the Washington Office were the supervision of trainees for the GM Patent Section and investigation work on GM products destined for production.

Mr. DeMott was a graduate of Syracuse University, receiving the B.S. degree in 1919. Following graduation, he was employed at the United States Patent Office in Washington, and remained there until 1926. During this period he continued his formal education at the National University Law School where he earned the degrees of LL.B. (1923), LL.M. (1924), and Master of Patent Law (1924).

His career, spanning a 40-year period, was spent entirely in patent law activities. This included 16 years as a patent attorney with a private firm in Washington, D.C., and seven years in his own firm prior to joining General Motors.

ROBERT L. EVERETT, co-contributor of "New Studies Provide More Information on Engine Rumble—A Phenomenon of High Compression Ratio Engines," is a research engineer with the Fuels and Lubricants Department, GM Research Laboratories. He is presently engaged in studies of air-fuel mixture distribution in multi-cylinder engines. His past projects have included the study of engine rumble as an abnormal combustion phenomenon and the development of instrumentation to detect abnormal combustion in engines during road test work.

Mr. Everett joined the Research Laboratories in 1957 shortly after receiving the B.S.M.E. degree from the Uni-

versity of Pittsburgh. He previously was granted a B.S. degree in education from this same university in 1951. He is a member of the Society of Automotive Engineers and Pi Tau Sigma, mechanical engineering honorary society.

ELWOOD K. HARRIS,

faculty member in charge for the typical General Motors Institute classroom problem, "Determine the Entropy Values for Low Pressure, Superheated Water Vapor," and the solution appearing in this issue, is chairman of the Product Engineering Department at G.M.I. In this position he is responsible for correlating departmental assignments in the general areas of engineering drawing, descriptive geometry, thermo-dynamics, fluid mechanics, and chassis and transmission design. He also teaches internal combustion engine courses in applied design.

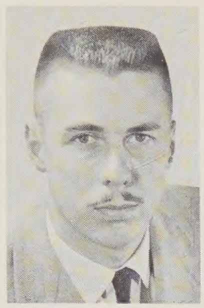
Mr. Harris joined GM in 1929 as an instructor in the Engineering Drawing Department at G.M.I. In 1940 he was made acting department head of the Product Engineering Department. He assumed his present position in 1946.

Michigan State University granted Mr. Harris the B.S.M.E. degree in 1929 and the M.E. degree in 1946. He is a member of the Society of Automotive Engineers and serves on various committees of this Society.

LEONARD R. HOSTETTER,

contributor of "Planning an Improved Electrical Distribution System for an Automobile Assembly Plant," is a process and production engineer in the Plant Engineering Department, Linden (New Jersey) Plant, Buick-Oldsmobile-Pontiac Assembly Division. His duties include layout of equipment and facilities for production operations and for model changeover rearrangements, design of electrical and mechanical equipment, and supervision of field installations.

Mr. Hostetter joined General Motors in 1951 as a General Motors Institute co-op student sponsored by Buick-Oldsmobile-Pontiac Assembly Division. He received the Bachelor of Mechanical Engineering degree from G.M.I. in 1956 after completion of his Fifth-Year Project. His Thesis—one of the requirements for this degree—dealt with the revision of the electrical distribution system at the Linden Plant, of which he writes in this issue. At G.M.I., Mr. Hostetter was elected to Alpha Tau Iota, engineering honorary society.



HARVEY J. MEEUSEN,

contributor of "The Application of Nomograms to the Solution of an Engineering Problem," is a junior project engineer in the Experimental Development Group at Diesel Equipment Division. His present responsibilities include theoretical analysis work and new product development.

Mr. Meeusen joined Diesel Equipment in 1953 as a learner detailer in the Product Engineering Department. A short time later he entered General Motors Institute as a co-op student. In 1958 he received the B.M.E. degree from G.M.I. His Fifth-Year Project Study was concerned with a theoretical analysis of the hydraulic valve lifter in relation to valve gear design.

Mr. Meeusen is a member of the Society of Automotive Engineers. He also is a member of the Cam and Valve Gear Subcommittee of the GM Committee on Engineering Computations.

JOHN L. PERAMPLE,

co-contributor of "Design of a Final Calibration Stand for a Pendulous Integrating Gyroscope," is a project engineer in the Test Engineering Department of the AC Spark Plug Division, Milwaukee, Wisconsin. He joined this Division in 1951 as a junior engineer after graduating from Marquette University with the B.S.E.E. degree. He was promoted to his present position in 1955.

Mr. Perample's work includes the design of experimental gyroscope and accelerometer test equipment for evaluating new and improved gyroscope designs. He also has been concerned with the design of production testing equipment for gun-bomb-rocket sights and testing equipment for a bombing navigational computer.

ROBERT P. ROHDE,

contributor of "Evolution and Design of Compact, High Performance Pumps for Power Steering Systems," is a senior project engineer in the Product Engineering Department of Saginaw Steering Gear Division. He is responsible for the design and development of power steering pumps and linkage booster power steering systems.

After graduating from the University of Michigan in 1950 with a B.S.M.E. degree, Mr. Rohde joined Saginaw Steering Gear Division as a college graduate in training in the Product Engineering Department. He entered the U.S. Army in December of that year and returned to Saginaw Steering Gear in January, 1953, as a junior designer working on the design and development of power steering systems. He was promoted to project engineer in September, 1953, and assumed his present position in 1956.

National honorary societies to which Mr. Rohde has been elected include Phi Eta Sigma (freshmen), Pi Tau Sigma (mechanical engineering), and Tau Beta Pi (engineering). He is a member of the Society of Automotive Engineers and has served on its membership, arrangements, program, and publicity committees.

RUSSELL F. STEBAR,

co-contributor of "New Studies Provide More Information on Engine Rumble—A Phenomenon of High Compression Ratio Engines," is a research engineer with the Fuels and Lubricants Department, GM Research Laboratories. His present work is concerned with studies of abnormal combustion in multi-cylinder engines.

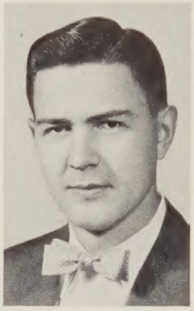
Mr. Stebar joined the Research Laboratories in 1954, shortly after receiving

an M.S. degree from Virginia Polytechnic Institute. His previous projects have included studies of surface ignition, knock, and hot starting in single and multi-cylinder engines. His experience also includes high speed motion picture photography of combustion in engines.

He is a member of the Society of Automotive Engineers and Pi Tau Sigma and Tau Beta Pi, honorary societies.

GEORGE E. SONNTAG, co-contributor of "Design of a Final Calibration Stand for a Pendulous Integrating Gyroscope," is a senior project engineer in the Test Engineering Department of the AC Spark Plug Division, Milwaukee, Wisconsin. He is responsible for the design of mechanical test equipment necessary for gyroscope and accelerometer production.

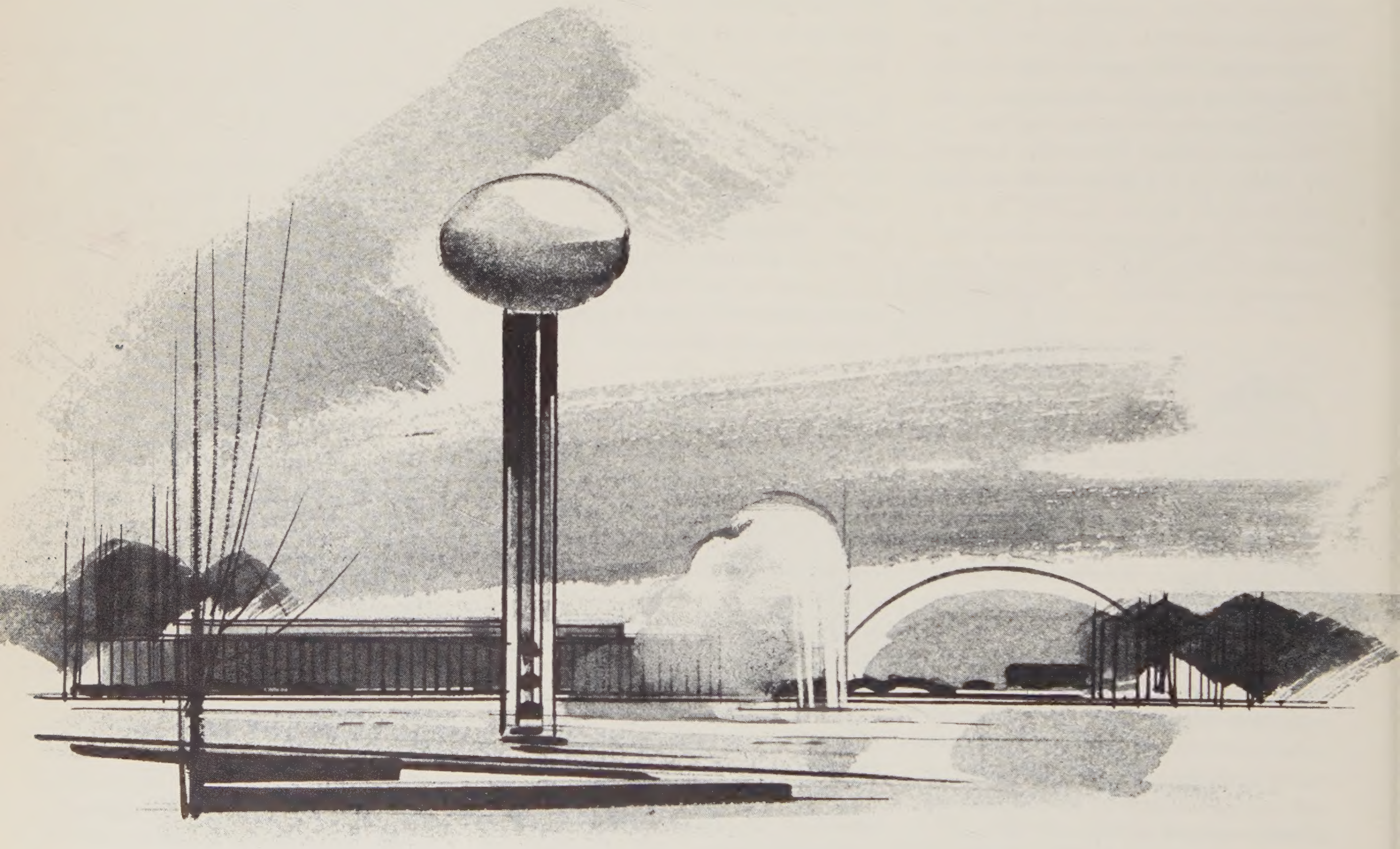
Mr. Sonntag joined AC Spark Plug in 1956 as a project engineer and was promoted to his present position on January 1, 1959. He has had previous experience with the Vapor Blast Manufacturing Company and the A. O. Smith Corporation. He was granted the B.S.M.E. degree in 1947 by Marquette University where he was elected to Tau Beta Pi, honorary society. His technical affiliations include membership in the National Society of Professional Engineers.



WARREN M. WIESE, co-contributor of "New Studies Provide More Information on Engine Rumble—A Phenomenon of High Compression Ratio Engines," is a senior research engineer with the Fuels and Lubricants Department, GM Research Laboratories.

Mr. Wiese received the B.S.M.E. degree from the University of Minnesota in 1950 and the M.S. degree in 1952. A short time later he joined the GM Research Laboratories as a junior engineer. Included in his previous projects was the development of an extension to the octane scale, which was tentatively adopted by the American Society for Testing Materials. His current projects deal with the investigation of abnormal combustion in high compression ratio engines, which includes the study of spark knock, surface ignition, engine rumble, and the deposit ignition resistance of fuels. He also has charge of all Fuels and Lubricants Departmental road test activities at the GM Proving Ground and supervises octane requirement and air-fuel ratio tests at the Proving Ground for GM car Divisions.

Mr. Wiese is a member of the S.A.E. and the Coordinating Research Council. He received the S.A.E.'s Russell S. Springer Award in 1959. He also is a member of Pi Tau Sigma and Tau Beta Pi, honorary societies.



ENGINEERING

ASSIGNMENT IN GM

An engine-transmission structure must be designed to have the required rigidity and also must be mounted in such a manner that the vibratory forces transmitted from the structure to the car assembly are at a minimum and at frequencies which will not produce sympathetic vibration of other components.

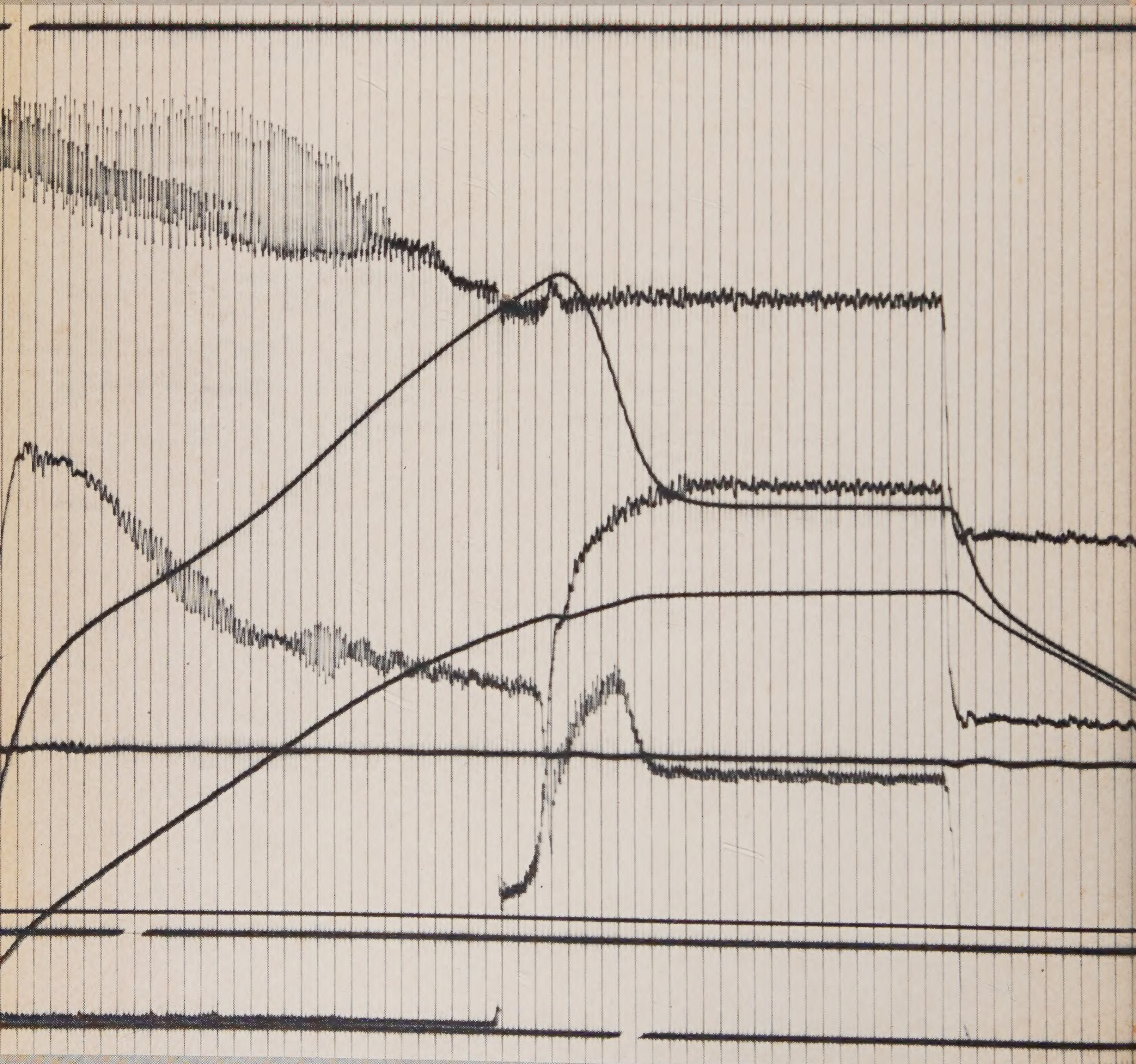
To determine the vibratory forces present in such a structure at the Oldsmobile Division, a response curve of displacement versus frequency is first obtained in both the vertical and horizontal planes. From this curve, the fundamental frequency of vibration of the structure is then determined. Next, engineers perform a "shake test" on the structure, as shown in the photograph above, to find the points of minimum and maximum vibration amplitude. The engine-transmission structure, suspended by Nylon ropes, is excited at the fundamental frequency of vibration by an electro-magnet. The structure is then probed to obtain information for plotting curves of displacement versus probe location. From the data obtained, points of maximum bending and minimum displacement are located and structural or mount designs changed as required.

In the photograph, James J. Henesy (left), dynamometer test operator, uses a vibration pickup to probe the structure. This pickup transmits an amplified vibration signal to an oscilloscope, being read by Bruce W. Trudgen, project engineer.

Mr. Henesy joined Oldsmobile in 1959 as a dynamometer test operator in the Engine Development Laboratory following his graduation from Western Michigan University with a bachelor of science degree.

Mr. Trudgen joined Oldsmobile in 1957 as a junior project engineer in the Experimental Test Laboratory and was promoted to his present position in September, 1959. He graduated from Michigan State University in 1954 with a bachelor of science degree.

GENERAL MOTORS ENGINEERING JOURNAL



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